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BLADE TIP GAP EFFECTS IN TURBOMACHINES - A REVIEW

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A review is presented of experiments examining the effects of blade tip gaps encountered in turbomachines and the methods by which the synthesised data are currently used in turbomachinery design and performance analysis. Data gained since the 1930's is subdivided for convenience into diffusing, or compressor-type flows, and accelerating, or turbine-type flows. There is a further sub-division into cascade and rotating machinery data.

The overall trend universally is that an increasing tip gap whose effect can reach over most or all of the blade height brings reduced overall performance of a turbomachine. Turbine data are in general more regular than the body of compressor data, possibly due in part to the enhanced effect of, usually, undefined boundary layers in diffusing flow. There is some evidence among the compressor and compressor cascade data that an optimum gap exists when the opposing effects of secondary flows, and tip leakage with rotor/wall relative movement tend to balance.

Comment is made upon the prediction and design models published in the literature.

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

A	blade aspect ratio = Z/c
C_h	stalling static pressure rise coefficient, defined in text
C_L	lift coefficient
C_{L_r}	lift coefficient retained at the blade tip section
C_m	mass flow coefficient = $\dfrac{\int_0^\lambda v\,dz}{\left(\dfrac{2(P_2-P_1)}{\rho}\right)^{\frac{1}{2}}\lambda}$
C_N	normal force coefficient
C_p	static pressure rise coefficient
$C_{\underline{p}}$	specific heat at constant pressure
c	blade chord
D	rotor tip diameter
E_L	blade end coefficient = $\dfrac{2(P_L-P_u)}{\rho U^2}$
F_y	tangential force on a blade row per unit span $= 2\pi r\rho V_x \left(\dfrac{r_{Z+\Delta}V_{x_{Z+\Delta}} - r_Z V_{x_Z}}{r}\right)$
$\underline{f}_N$	frequency
g	staggered spacing of blades = $S\cos\xi$
g_0	gravitational constant
J	Joules equivalent
K	measured lift coefficient retained at blade tip = $C_{L_r}/C_{L_{2D}}$
$K_{1,2,3}$	constants
k	cavitation number

N	rotational speed
n	number of blades in a blade row
P	stagnation pressure
P_L	stagnation pressure at blade pressure surface
P_u	stagnation pressure at blade suction surface
p	static pressure
q	dynamic head
R	tip reaction
RBH	reduced blade height configuration
RC	recessed blade casing configuration
r	radial dimension
S	blade pitch
SH	shrouded blade configuration
T	stagnation temperature
t	static temperature
t	profile maximum thickness at the tip section
U	blade speed
V	flow velocity
x	axial co-ordinate
y	pitchwise co-ordinate
Z	blade span
z	spanwise co-ordinate
α_1	flow entry angle to blade row
α_2	flow exit angle from blade row
α_M	mean air angle $= \tan^{-1}\left(\frac{\tan \alpha_1 + \tan \alpha_2}{2}\right)$

Γ_m	mean local value of circulation
δ^*	boundary layer displacement thickness
γ	ratio of specific heats
η_{AD}	adiabatic efficiency of the stage
λ	tip gap height in turbomachine and cascade or tip gap semi-height in imaged cascade experiments
υ	tangential force thickness of boundary-layer
ξ	blade stagger angle
ρ	density of working fluid
σ	throttling coefficient = $(V_a/U)^2 \Big/ \frac{2(P_{out} - P_{in})}{\rho U^2}$
ϕ	mass flow coefficient = V_a/U
ψ	work coefficient = $\Delta H/U^2$
ζ	incidence angle

Subscripts

2D	two-dimensional flow
x	axial direction
1	entry plane of blade row
2	exit plane of blade row
a	axial component
out	at compressor outlet
in	at compressor inlet
t	tip value
h	hub value
max	maximum value
GAP	inter-blade row gap value
R	Rotor
S	Stator

Superscripts

~ free stream quantities

— mean value

1. INTRODUCTION

In order to gain high cycle thermal efficiency, gas turbine engines must use high compressor pressure ratios, resulting in relatively small annulus height in the high pressure section of the compressor. In this region the tip gap, which may be of fixed dimension, becomes large as a percentage of a characteristic blade dimension. Since efficiency loss through the effects of blade tip clearance is a recognized feature of compressor operation, increasing generally with non-dimensionalised tip-clearance, it might be anticipated that tip loss would have an increasingly dominant effect. This trend runs, unfortunately, counter to the performance and geometric requirements of new generations of engine and particularly the concept of the energy efficient engine.

The layouts proposed for future engines show the turbomachinery to occupy an increasing proportion of the total volume and hence total mass. Relating crudely the cost of manufacture to the total mass the costing of components may be apportioned in relation to their proportion of the total engine mass. The turbomachinery thus represents a very significant cost item in the manufacture of a gas turbine and, although the turbine demands the costlier materials, the compressor, because of its relative size, is at least as expensive in manufacture. Costs would however be reduced if manufacturing tolerances could be relaxed and so, from the economic point of view, increased tip clearance would be advantageous.

In addition to the requirements for the more efficient use of the available fuel and the maintenance of reasonable manufacturing costs, the designer of the gas turbine engine must maintain high standards of safety in operation. As a consequence, the equilibrium running line on the compressor characteristic map must be sufficiently separated from the stability limit line to maintain an adequate surge margin, permitting normal transient operation without destabilising the system. The stability limit of a compressor is governed by the onset of rotating stall which, depending upon the system dynamics, may develop into a surge situation. Rotating stall is itself usually identified both for part-span and full span stall at the outer sections of the blading, the region of the rotor tip gap, so it may be anticipated that both the initiation of the rotating stall and its characteristics may be affected by interaction with the tip gap aerodynamics - such reasoning has led to the introduction of tip treatments in certain compressor designs.

It is seen then that the tip gap aerodynamics plays an important part in the performance of a compressor and will be of increasing significance with new engine designs. Historically though, the record of dealing with the tip gap problem is depressing. Osborne Reynolds, who held what was possibly the first patent on a turbomachine, abandoned his work when he found that only low efficiency could be obtained on a water turbine of small size that he designed. This inefficiency was due, in the main, to tip loss, which would have become less dominant for a gap of the same dimension in large size turbomachinery, a point not pursued by Reynolds. In a discussion on axial flow pumps and impellers held at the Institution of Mechanical Engineers in 1956 Professor A.D.S. Carter suggested a 'rough and ready rule' that clearances up to about 2% of the blade height were assumed to have practically no effect. The advances from the time of Osborne Reynolds were then mainly in recognising that a tip gap could have an effect, as yet poorly quantified.

Although every turbomachine manufactured must, by definition, contain some form of gap at the blade tip it is surprising that from the time of Reynolds to the present so little research has been done in this field. Much of the work, especially on compressors, has tended to be piece-meal and many quantities which may now be recognised as important, for instance the annulus wall boundary-layers, have often not even been measured. Some data are however available for cascades and rotating machinery, both compressive and expansive and it is these data which have been reviewed here.

As well as indicating what has been accomplished, this document exposes those areas necessary, in the light of new engine design, for future investigation. The review presented is thus seen to be a timely synthesis of data measured over a number of years yet not collated hitherto in this manner.

2. BACKGROUND

There is evidence to suggest that increasing the tip gap of a compressor rotor involves a significant efficiency penalty. Enlarging the gap by 1% of the blade height may be expected to yield approximately a 2% drop in compressor efficiency in certain current large scale gas turbines using blades of large dimension. Towards the rear of a high pressure compressor, where blades are of small dimension, the effect could be greater. With the need for increased component efficiency in engines, this represents a severe penalty.

The mechanics of the loss inducing flows associated with the rotor tip region remain largely not understood so that attempts to model the flow or reduce its effect are thwarted. It is known however that small geometric variations in the region of the rotor tip can have a substantial effect upon performance; not only do increased tip clearances generally reduce efficiency but certain types of tip treatment are known to improve the compressor behaviour close to the stability limit line and hence improve the surge margin (1). At the same time there may be an efficiency penalty in compensating for the increased operational range. As with the case of a tip gap interacting with a rotor blade and a plain annulus wall, the detailed aerodynamics associated with various types of tip treatment, and thus the reasons for changes in compressor overall performance, are not known.

Because of the known effect of tip treatments it may be reckoned that the stability limit of a compressor, marked by the onset of rotating stall is likely to be governed to some extent by the tip gap geometry and aerodynamics. Similarly the post-stall recovery, which because of hysteresis effects in the process, is unlikely to occur at the same point on the characteristic and is also likely to be affected by the flow behaviour at the tip gap. Stability limit operation and post-stall recovery represent a continuing area of vital interest to the manufacturer and user.

Rotating stall represents one class of non-axisymmetric flow and although a compressor is designed for uniform inlet flow conditions, a second class of non-uniform flow, circumferential distortion, may be present in application. The gas turbine when installed often has its inlet flow distorted by interactions with upstream bends, struts, shock systems or, for an aircraft undergoing a high 'g' manoeuvre, boundary layer separation at one side of an intake or partial masking of an intake by the aircraft body. Such distortions although predominantly quasi-steady and hence planar usually have some time-unsteady content. There is much evidence to show that the circumferential planar distortions generated by such features have a deleterious effect upon performance altering the shape of the running line and precipitating compressor destabilisation by depressing the stability limit line to the compressor operating point (2). Although this form of distortion is time-steady in the absolute frame of reference, relative to the rotor it is seen as a time-unsteady effect, producing a time-unsteady reaction from the rotor blade (3). The time dependent features include a complex hysteresis loop in the lift and incidence line (3), but can also involve rotating stall which may be initiated by the distortion. Since it has already been observed that the characteristics of the rotating stall may be dependent upon the rotor tip gap it may be anticipated that an interaction would occur between the distortion generated and the tip gap, particularly under low flow/high blade load conditions.

While it is possible to make the general observation that reduction in tip gap size improves the efficiency of the unit it is recognised that higher costs are involved. It is a requirement for the engineer to know the trade-off between compressor performance as a function of rotor tip clearance and cost in order that the economics of manufacture may be measured against cycle efficiency.

The effects of the tip gap in a turbine parallel largely those of a compressor. A loss of efficiency with an associated loss in work coefficient is involved and the perturbed aerodynamics may reach over much of the blade height. In the case of a turbine though, de-stabilisation is not of interest but the tip gap problem is enhanced because of the sizes of gap necessary to accomodate thermal growth.

The region of the rotor tip then represents a sensitive area which possibly affects the compressor operational efficiency, stability limit, post-stall recovery, distorted inflow performance and turbine efficiency, flow coefficient and detailed aerodynamics. To make significant advances in these areas resulting in reduced tip losses through geometric variations, casing treatment and tip feathering calls for a realistic flow model gained from an understanding of the physics involved. The flow is however highly three-dimensional, viscous, time-dependent and possibly compressible; interdependent effects that cannot be modelled within the state of today's art. To obtain the data upon which a model, resulting in a designers and analysts code, may be based, it is necessary to resort to experiment. Such experiment though needs to be comprehensive in its scope and detailed in its measurement.

3. BASIC FLOW MECHANISMS IN TIP GAPS

A substantial body of evidence exists to indicate that while a turbomachine may be manufactured with less expense if it has large tip clearances, there is a progressive performance penalty with increasing tip gap. The mechanics of the flow that result in performance loss are not understood in detail, but it may be assumed that the contributory factors include:

a. the pressure difference between the suction and pressure surfaces of a blade, resulting in a leakage flow through the tip gap from the pressure to the suction surface. In the case of an aircraft wing this results in the tip vortex and the progressive loss in lift of the wing section towards the wing tip. Associated with this in a compressor is the vortex flow and related radial distribution of loading to which the blading was designed.

b. the presence of the boundary layer on the casing or hub wall of the turbomachine. This shear flow itself affects the tip region performance encouraging secondary distributed vorticity which, locked in the region of the interface between the blade suction surface and the annulus wall, results in a three-dimensional separation contributing to loss of efficiency.

c. the relative movement between the blade and the boundary layer on the casing or hub wall, encouraging a jet flow through the gap. For a compressor blade the jet flow relative to the blade is from the pressure to the suction surface, additive to the flow mentioned in 'a'. For a turbine blade, because the relative movement is in the opposite direction, the jet flow relative to the blade is from the suction to the pressure surface, in opposition to the mechanism of 'a'.

d. the size of the gap. Clearly, as the gap increases in size, resistance to the flow mechanisms mentioned in 'a' and 'c' is reduced, so their effect is generally increased. For very large gaps, the effect of reduced blade height in reduced work associated with the change of blade height can also become noticeable.

Attempts have been made to understand in detail the individual effects listed above, possibly with a view to superimposition of results to gain an understanding of the complete phenomenon.

Of the three fluid mechanical, as opposed to geometric, features listed, the flow at the tip due to the pressure difference between the surfaces may be considered inviscid, while those due both to the presence of the casing boundary layer and the relative motion between the rotor tip and the casing are viscous. It is possible to separate these effects somewhat so that the inviscid and viscous effects may be examined in isolation and this has been an approach used by various workers.

4.0. COMPRESSOR CASCADE EXPERIMENTS

4.1. Tip Gap Behaviour with Inviscid Flow

A technique to investigate the inviscid flow through a tip gap involves the use of imaging - a blade in cascade split at the mid-height section to produce effectively a gap between two blade ends. Ignoring the viscous growth on the end surface of the blades and that on the blade sections themselves, for a uniform inlet flow such as one would be likely to have at a cascade mid-region, the flow may be reckoned to be inviscid. In a two-dimensional cascade, the two tip sections resulting from the geometric split are identical resulting in identical, image flow at each tip. There is thus a line of symmetry at the mid-gap height which becomes the height of the pseudo-inviscid gap to be investigated. This geometry was used by Khabbaz (4), Yokoyama (5) and Lakshminarayana and Horlock (6, 7).

The focus of attention in Khabbaz's (4) experiments was in determining the effect of the gap in inviscid flow upon the stall characteristics of the blade section. He noted that, at the tip section, stall was delayed to a higher flow incidence in the presence of a gap than without the gap. He also found that increasing gap size increased the blade loading near the tip while, for the remainder of the blade, conditions remained substantially unaltered. Yokoyama (5), using the same test facility observed that, while the lift coefficient increased near the tip region the axial velocity and the turning angle locally decreased (figure 1). These observations discredited a momentum theory model proposed by Khabbaz but led to the conclusion that the tip vortex made a strong contribution. Yokoyama also observed that the enhanced lift close to the tip was similar to that measured by Holme (8) on a rectangular wing of unity aspect ratio (figure 2). Yokoyama's experiments were all conducted at gap semi-height/chord ratio λ/c of 0.03 while Khabbaz varied the gap from $0 < \lambda/c < .10154$. Lakshminarayna and Horlock (6, 7) using a similar geometry, measured the spanwise lift distribution varying the gap from $0 < \lambda/c < .426$. At small gap geometry ($\lambda/c < .03$) the results were qualitatively similar to those of Yokoyama, a modest increase in lift resulting in the tip region (figure 3). With larger gaps, the blade experienced a reduction in total lift, the spanwise extent and overall lift reduction being progressive with gap size. In terms of normal force coefficient variation with blade span and gap size, their data were closely co-incident with Yokoyama. It was, however, observed (7) that in the range of gap size normally employed in turbomachines, there was always a slight increase in average normal force coefficient (figure 4).

Lakshminarayana and Horlock also measured a progressive increase in induced drag with increased gap size (figure 5). This change with gap/chord ratio inferred a change in shed vorticity at the tip, leading to the conclusion that, in the experiment with imaged blades separated by a gap, not all of the bound vorticity was shed at the blade tip. That towards the blade leading edge bridged the gap. This was in contradiction to the supposition of Dean (9) who stated that in an inviscid flow field, as soon as the tip clearance becomes finite, no vortex lines could cross the clearance gap. Dean concluded that all of the bound vorticity must then trail off as a vortex sheet of varying strength behind the blade in the manner of a wing. Such a model which supposes that the flow field comprises a series of insulated, two-dimensional lamellae laying in the x, z plane, must admit the presence in the flow of infinite values for $\frac{\partial v}{\partial y}$ and $\frac{\partial p}{\partial y}$, the rates of change respectively of velocity and pressure in the spanwise direction. Even in an inviscid fluid, the proximity of an aerofoil section promotes non-infinite pressure gradients around the aerofoil tip and hence, for the geometry under consideration, within the tip gap, resulting in the transverse flow that comprises the tip vortex. It may then be concluded that the resulting pressure field within any particular lamella (x, z plane) embracing the gap region promotes a varying velocity producing circulation and hence vorticity in the x, z plane. Extending the argument in an inviscid fluid to a cascade wall, as long as a non-constant pressure field exists at the wall, vorticity will enter the wall.

This reasoning supports the observations of Lakshminarayana and Horlock (6) for an inviscid flow field. They developed the concept of vortex lines, which originally passed as bound vortices from a blade to its image at zero tip spacing, being progressively shed as the gap increased. From the above reasoning it may be concluded that all of the bound vorticity would be shed at the moment when the mid-gap plane (x, z) representing the pseudo-wall no longer contained any static pressure variations.

For low gap/chord ratios, the experiments led to a model based upon the observation that the shed vorticity from the blade tip was $(1-K)\Gamma_m$ where K, the measured lift coefficient retained at the tip as a proportion of its two-dimensional value, was derived from experiment (figure 6) and Γ_m was the mean local value of circulation.

With very large values of gap/chord ratio, the tip aerodynamics tended, with reducing interaction between the two blades, to those of an isolated aerofoil where all the tip region-bound vorticity was shed in the tip region ($C_{L_r} \to 0$). The shed vorticity thus tended to Γ_m, the mean local value of wing circulation.

4.2. Simulation of an Inlet Shear Flow

In a further series of experiments Lakshminarayana and Horlock (6, 7) allowed for the presence of a wall and shear flow in three ways, by placing

a) in the gap mid-span a splitter wall extending from the blade leading to trailing edge. (6)
b) upstream of the gap, a perforated splitter plate whose wake was convected through the gap. (7)

c) a downstream extension of the perforated splitter plate to continue it through the gap. (7)

The gap was at all times variable from zero size.

All of these geometries led to the generation of secondary flow as a consequence of the attendant shear flows. In case (a), no shear was present at the cascade entry plane and only built up as a thin boundary layer through the cascade channel. The results indicated a reduction in total blade lift and increase in total induced drag coefficient, commensurate with breaking the bound vorticity previously bridging the gap between the blade and its image.

The effect of convecting through the test section an upstream generated shear flow whose semi-span was about five times the gap semi-height was that, within the channel, a secondary flow was generated. This was opposite in direction to the leakage flow, observed in the test with uniform inlet flow, which it met to form a core of heavy loss at the exit plane in the mid-passage. With no gap, this loss was locked into the corner created by the interface between the blade suction surface and the sidewall as measured by Peacock (10). The spanwise distribution of loss coefficient indicated that the effect of the secondary flow was to spread the loss coefficient over a larger part of the blade span and the magnitude of loss increased with gap size. Over much of the blade span the normal force coefficient based upon local inlet dynamic head was reduced somewhat, except in the region of the tip where it was enhanced. The effect of closing the gap was to eliminate the reduction in normal force coefficient immediately at the tip section where the normal force coefficient reached a maximum. The effect of the secondary flow upon the air leaving angle was to reduce the underturning created by the leakage flow through the gap.

Experiments with geometry 'c' confirmed that the effect of a shear flow was to create a secondary flow structure operating in opposition to the leakage flow. It was noted that the two flows could be to some extent balanced by tuning the tip gap when the leakage flow through the gap as a result of the pressure difference across the blade from the pressure to the suction surface, reducing α_2 locally, was balanced by the effect of the secondary flow in the region of the sidewall generally increasing α_2.

At $\lambda/c = 0.04$ Lakshminarayana and Horlock saw that the leakage flow tended to displace the corner separation zone at the blade and end wall surfaces: also the secondary flow, in opposing the leakage flow, prevented the leakage flow from moving down the suction surface along the blade passage. The effects of leakage were confined to a region near the suction surface while the flow elsewhere was greatly influenced by secondary flow and separation. The effect of leakage was therefore to increase the underturning at the suction surface near the wall and decrease it in most other regions, improving the lift distribution along the blade.

4.3. Effect of Relative Movement Between a Blade and Wall

In balancing the effects of the pressure difference across the blade surfaces (inviscid) with the secondary flows (viscous) no account was taken of relative movement between the blade and the sidewall or viscous layer. In a normal situation for a rotating machine this effect is present, resulting in changes in the leakage flow rate at the gap because of convection of the viscous flow through the gap. This viscous effect is in opposition to the secondary flow viscous effect and in sympathy with the inviscid flow direction across the gap. It may then be concluded that the effect of relative movement could be significant.

In order to investigate this Gearhart (11) constructed a facility in which air passed from a plenum chamber through a bellmouth and entry region to a test section containing a blade with tip clearance. Along the entry region and across the blade tip an endless belt was designed to move in the direction of the airflow. Tests investigated the effects both of relative velocity and blade end geometry. Variables investigated in the programme included gap height, blade end geometry and a blade end coefficient (E_L) defined by

$$E_L = \frac{P_L - P_U}{\rho U^2/2}$$

relating the difference of the average stagnation pressure on suction and pressure surfaces of the blade to the dynamic pressure associated with the belt velocity. For zero belt speed, the value of E_L was ∞. For non-zero belt speed (kept constant in the programme) E_L was varied by altering the pressure difference across the blade and by varying plenum chamber pressure.

Although this work was aimed primarily at investigating cavitation problems in water pumps certain data of use to the aerodynamicist emerged since the experiments reported were in air. They were:

a) The effect of decreasing the tip gap was to increase the local maximum depression in the gap - a consequence of decreasing the effective throat in the tip region.

b) The effect of increasing the blade end coefficient, by increasing the pressure drop (P_L - P_U) across the gap was to reduce the absolute level of non-dimensional velocity (figure 7) and hence non-dimensional mass flow through the gap (figure 8). One test with zero belt speed ($E_L = \infty$) produced the minimum non-dimensionalised leakage flow.

It may be concluded that the effect of blade/wall relative velocity was to increase the tip leakage flow, the effect of viscous flow transported at the wall being in sympathy with the leakage flow normally present in a stationary experiment.

c) Tests in which the blade end geometries varied yielded sharply differing results.

The experimental programme, while accounting for relative movement between blade and sidewall did so with a belt moving in the flow direction. The boundary layer at the sidewall remained two-dimensional with respect to the blade, which does not represent the situation encountered in a turbomachine where the boundary layer is highly skewed with respect to a rotor blade.

Dean (9) also executed a moving belt experiment using a rectilinear cascade in which one wall contained a moving belt adjacent to a gap at the end of the cascade aerofoils. The belt covered an axial distance from 0.4 blade chords upstream of the cascade inlet to the cascade exit plane. In Dean's experiment the blades, inlet flow and belt movement were in the correct relative relationship: the boundary layer was not convected simply in the direction of the stream flow, but had a component in the pitchwise direction of the blading. Further, this pitchwise component being superimposed because of its viscous properties and being most evident at the belt surface, imparted a skew to the boundary layer as it would be seen by the rotor blade of a compressor.

Initial experiments with the belt stationary and with no tip gap confirmed the observations of Peacock (10) and Lakshminarayana and Horlock (7) that a region of low energy fluid was present at the blade suction surface in the tip region. This phenomenon is also present in a compressor and detailed measurements of it have been made by, among others, Dring, Joslyn and Hardin (12, 13). Upon creating a finite gap, Dean noted that this region was moved by tip leakage away from the suction surface to a position approaching mid-pitch close to the cascade sidewall. This also was in line with the measurements of Lakshminarayana and Horlock. It was further observed that this effect was enhanced by operating the belt when the viscous effects of the flow were additive to the tip leakage flow. It was however noted that a tip gap (λ/C) greater than 3.4% did not materially affect the position of the low energy core in the cascade channel and a wall speed greater than 127% of the flow axial velocity also had only a small effect. Qualitatively the effect of wall speed was dissimilar to that of Gearhart (11) who found that the velocity and mass flow through the gap increased with belt velocity (reduced E_L) the effect becoming more dominant with higher belt speeds. It would then be anticipated that the low energy core measured by Dean would have been moved progressively farther across the passage.

5.0. EXPERIMENTS IN COMPRESSORS

Because of the historical lack of suitable high response rate instrumentation and data logging facilities most measurements on rotating machinery have not been made to the same detailed standard as on cascades. In general the effect of tip gap variation upon overall performance parameters, work coefficient and efficiency, has been examined. In experiments using water as a working medium such measurements have been supplemented by visual observations and in certain experiments radial traverses have been made. Currently proposed research programmes are however capable of exploiting modern instrumentation for detailed flow evaluation.

5.1. Effect of Tip Gap on Overall Performance Parameters

One of the earliest experiments in which the tip gap was varied in a controlled manner was that of Ruden (14) using a single stage fan. His data established the shape of that to follow - an almost linear relationship showing both efficiency and work coefficient to reduce with increased tip gap. Six tip gaps were employed from $.002 < \frac{\lambda}{D} < .012$ and the operational characteristics in terms of work coefficient and efficiency were progressively depressed with increase in tip gap (figure 9). The observed linear relationship between efficiency and work coefficient with tip gap respectively resulted from cross plotting these data on lines of constant mass flow function (figure 10).

Quoting results from an industrial compressor De Haller (15) recorded an almost linear drop in peak efficiency from 93% to 84% as the tip gap was quadrupled and Fichert (16) found a 3% fall in efficiency of a machine when the radial clearance was doubled.

Much of this earlier work covered a range of tip gap size larger than that normally employed in current designs, but the data did serve to establish trends which have continued to be confirmed with one or two notable exceptions. The rate of efficiency loss was generally from 1.25 - 1.50% per .010" increase in gap size.

An investigation on a three stage axial water pump was pursued at design point operation by Rains (17). Tip clearance, which was controlled by shims fitted beneath the rotor blade root was altered from .007" to .0357", a dimensionless range of 24% of the blade tip section maximum thickness or 1.4% of the tip chord. Efficiency fell by 3.5% (figure 11) and the work coefficient, defined as the work input ratiod to the dynamic pressure associated with the tip speed, fell by .007 (figure 12), a slightly higher rate than that noted by Ruden who measured the same loss over a larger range of tip gap.

Rains' work, in which water was the working fluid, was however aimed mainly at investigating the phenomenon of cavitation. Since cavitation was formed where the fluid pressure dropped to the vapour pressure and the region of lowest pressure was the core of the tip generated trailing vortex, it had been established (18) that the inception point of cavitation was in the tip region and not on the blade surface as traditionally supposed. Rains' observations have value in the gas compressor field. He noted that for very large tip clearances the geometry of the vortex was similar both for rotor and stator. With small tip clearances for a stationary blade the origin of the vortex was just behind the quarter-chord position, an observation that was closely aligned with that of Lakshminarayana and Horlock (7) who used this as geometric input for a subsequent model. With rotating blades however, Rains observed that the vortex was locked to the leading edge of the blade profile. Defining a parameter k, the cavitation number, which can be understood to increase as the cavitation performance decreases, he noted that, instead of a linear variation of k with tip clearance there was a region of small tip clearance which showed a marked decrease in performance, possibly the first indication of non-linearities resulting at small tip clearance.

Rains also noted that the tip clearance flow was appreciably increased as a result of the rotor/wall relative velocity, confirming the cascade work of Gearhart (11) and Dean (9).

Rains' investigations were extended by Williams (19) to investigate the off-design operation of the same pump, but in single stage configuration. While most of his work was at constant rotational speed, the mass flow was varied and, while it approached stall, did not enter it. The range of tip clearance was selected to bracket that used in turbomachinery practice, $.00368 < \frac{\lambda}{c} < .03742$. There was some scatter in Williams' results, but he found that the pressure coefficient and the efficiency decreased approximately linearly with gap size, the maximum rate of change being at the lowest flow rate used corresponding to the highest pressure coefficient (figure 13). Cross plotted in terms of efficiency v flow coefficient a series of well recognised curves with a maximum efficiency at the design mass flow resulted, the curves being depressed progressively with increased tip clearance, a confirmation of the trends observed first by Ruden (14).

In addition to their own results (discussed later), Jefferson and Turner (20) also re-presented some earlier results of Bowden and Jefferson (21) from an eight stage axial compressor. Over the range of tip gap which varied from .009" to .054" ($.0144 < \frac{\lambda}{c} < .0864$) there was an almost linear drop in peak efficiency and a fall in temperature rise coefficient (figure 14) whose rate increased progressively with tip clearance. Although their commentary does not mention it, the data suggest an initial improvement in stability limit line with subsequent slight decay.

5.2. Radial Survey Data

Williams (19) conducted surveys of axial velocity behind the rotor row. These indicated a velocity decrement towards the tip region in keeping with the inlet boundary layer profile.

At flow rates above the design value there was little change of velocity with variation in tip gap, but at flow rates less than the design value the effect of the largest tip gap was marked over the outer 25% of the blade height. It was however at high mass flows that the effect of high tip clearance reduced locally the work coefficient. These data are commensurate with reduced turning of the flow because of tip leakage, the effect covering the outer 25% of the blade height. Early work on the subject of tip clearance effects was reported by Hutton (22). Using a very low solidity fan he conducted tests at tip gaps of $\frac{\lambda}{Z}$ = 0.5, 1.5, 2.5, 3.5 and 4.5%. He noted that alteration of the tip gap led to substantial changes in the exit axial velocity (figure 15) and swirl angle over the whole of the blade height. The effect of tip gap was thus more global in its extent than that observed by Williams. In comparing the data from the two extreme gaps used, Hutton found that the effect on static pressure rise extended over all of the blade height leading to an almost linear decay of work coefficient with tip gap (figure 16).

5.3. Stability Limit Considerations

One important observation made by Hutton (22) and yet strangely not apparently pursued by subsequent workers, was the evident change in the compressor stability limit operation with tip gap (figure 17). Certain other work has identified the problem of surge margin erosion with tip gap size. Hutton found though that, in addition to the stability limit being encountered at higher mass flow coefficients with increase of tip gap, the strong hysteresis effect was removed. Recent experiments of Peacock and Das (23) supporting an examination made by Greitzer (24) confirm that the jumps associated with the hysteresis indicate changes in the rotating stall pattern. Hutton's measurements then indicate that the pattern of rotating stall may be governed by the tip gap geometry, a point not considered thus far either in rotating stall investigations or tip gap experiments.

Jefferson and Turner (20) pursued tests on an air compressor of six axial stages and low overall pressure ratio, using a range of blade designs and tip geometries. Most of their work was conducted with shrouds on the rotor and/or stator rows and the effects of varying shroud clearance were examined. In this case, clearance was defined as the axial distance of the shroud knife-edge and the compressor sealing face, so the geometry employed could not be equated with that under review. The conclusions nevertheless have qualitative value in that with increased clearance, yielding higher tip leakage, the efficiency at any particular mass flow coefficient fell giving a lower pressure rise coefficient. The stability limit of the compressor was however delayed to a lower flow coefficient, an observation that was in direct contradiction to that of Hutton (22). In comparative tests using consecutively shrouded blades and unshrouded blades with tip clearance they found that better performance resulted in having a tip clearance geometry and this was associated with suppression of a blade stall condition with the presence of the tip gap.

5.4. Tip Gap Optimisation

Spencer (25) used a four bladed axial flow water pump for his investigations which included tip clearance variation from 0.6 - 2.4% of the blade height. He found that the effect of tip clearance was the greatest at low flow rates when the pressure differences at the gap were at their greatest. The 1.8% increase in clearance (based on blade height) yielded a 15% drop in design flow and an overall drop in efficiency. There was an initial improvement in efficiency as the tip gap was opened (figure 18) and though, in his paper, Spencer thought the effect to be spurious, there were in the associated Communications observations by Desmur (26) who cited the work of Ténot

(27) and Daily (28), by Medici (29) and by Yamazaki (30) that there was an optimum tip gap for operation. Such observations were in line with that of Rains (17). Although all of their data were not consistent at small tip geometries Jefferson and Turner (20) concluded that a small tip gap, of the order of 1% of the blade height, would be beneficial in a design, a conclusion in sympathy with the communications (26, 29, 30) relating to references 22 and 25. This admitted the possibility of an optimum tip gap, indicated by Lakshminarayana and Horlock (6, 7) although in general not confirmed by the later main corpus of data available from the range of workers in the field.

The concept of an optimum tip gap was further pursued by Lakshminarayana (31) who postulated that, since the secondary distributed circulation and the tip leakage flow were mutual opposition at the tip, an optimum should exist when the sum of the two associated secondary circulations was zero. He found a qualitative agreement in applying this principle to the results of Dean (9) and Hubert (32).

5.5. Annulus Boundary Layer Considerations and Peak Pressure Rise

Based upon earlier results from a 12 stage compressor Smith (33) concluded that, by the fourth stage, essentially repeating conditions in terms of velocity profile had been reached. Using, in consequence, a four stage low speed compressor as the vehicle for his investigation and considering data from the third rotor, third stator and fourth rotor only, he conducted a comprehensive series of tests with a large range of geometric variables. Arguing that the staggered pitch was a better reference dimension than any other for non-dimensionalising parameters, he produced curves of constant non-dimensional tip clearance on a non-dimensional displacement thickness versus non-dimensional pressure rise field, both for the casing and hub regions of compressors (figures 19a and 19b). The data indicated the powerful effect of the magnitude of the compressor pressure rise upon the annulus displacement thickness for any tip gap. It also showed that the displacement thickness was strongly affected by the gap size, reduced gap size permitting more energisation of the wall boundary-layer to keep it thin. A further important conclusion of Smith's work was that stall was likely to occur at a limiting value of the boundary layer that depended upon tip clearance, indicating that the tip clearance contributed towards control of the limiting stall mechanism. In passing, it is noted that the tip gap alone was not enough to explain the variations of stalling pressure coefficient, but that the axial gap between blade rows also contributed, the lower the gap, the higher the stalling pressure rise at constant tip clearance. Because the data to hand did not cover a wide enough range of tip gap, the curves of constant tip clearance came "from engineering judgment seasoned by (appropriate) boundary layer data." Plotting data for three different blade geometries in terms of the peak static pressure coefficient against the tip clearance, Smith produced a series of straight lines indicating reducing performance with increasing tip gap (figure 20), thus adding a further confirmation to the work of Ruden, and others already cited.

Extending Smith's work and pursuing an interest in the stall pressure rise capabilities of compressors, Koch (34) identified tip gap as one of the factors influencing the stall pressure rise capability of multi-stage compressors. Varying tip clearances from 0.7% to 3.4% of the blade height by cutting the blades accordingly and using data at a non-dimensional gap of 0.055 (the normalising dimension being the average pitch-line gap) he showed that the non-dimensional static pressure rise coefficient reduced with increasing gap size and was insensitive to aspect ratio in the range $2.0 < A.R. < 50$. While the trend was similar to that encountered by other workers, the noteworthy features are that over the normal range of tip gap the results were not quite linear and that at small to zero tip clearance the pressure rise characteristic rose sharply (figure 21). It certainly contained no indication of an optimum tip gap.

To relate Koch's data to those of other workers it is necessary to understand the parameters in which he worked. Koch defined his stalling static pressure rise C_h as:

$$C_h = \frac{JC_p t_1 \left[(p_2/p_1)^{\frac{\gamma-1}{\gamma}} - 1 \right]_{Stage} - \frac{1}{2g_0}(U_2^2 - U_1^2)_{Rotor}}{\frac{1}{2g_0}\left(V_{1\,Rel\ Rotor}^2 + V_{1\,Rel\ Stator}^2 \right)}$$

Although nominated as a pressure rise coefficient this is really an enthalpy rise coefficient which, assuming that repeating stage conditions are fairly closely obeyed (see Smith (33)) so that the stagnation pressure ratio varies as the static pressure ratio, may conveniently be written

$$C_h = \frac{2gJC_p t_1 \eta_{AD} \left(\frac{t_2}{t_1} - 1 \right)_{Stage} - \left(U_2^2 - U_1^2 \right)_{Rotor}}{V_{1\,Rel\ Rotor}^2 + V_{1\,Rel\ Stator}^2}$$

Other work already reviewed has indicated a linear degradation of efficiency with tip gap. Jefferson and Turner (29) and Rains (17) have measured an almost linear degradation of work coefficient $\Delta T/\frac{1}{2}U_m^2$ with gap size.

Since $$\frac{\Delta T}{\frac{1}{2}U_m^2} = \frac{(T_2 - T_1)}{\frac{1}{2}U_m^2} = \frac{T_1\left(\frac{T_2}{T_1} - 1\right)}{\frac{1}{2}U_m^2} \simeq \frac{t_1\left(\frac{t_2}{t_1} - 1\right)}{\frac{1}{2}U_m^2}$$

for constant inlet conditions and blade speed, $\left(\frac{t_2}{t_1} - 1\right)$ it may be concluded, varies linearly with gap. We see therefore that

$$C_h = \frac{K_1 \eta(\lambda) \left(\frac{t_2}{t_1} - 1 \right)(\lambda) - K_2}{K_3}$$

where $K_{1,2,3}$ are constants, so

$$C_h = \frac{K_1 f(\lambda)^2 - K_2}{K_3}$$

which is quadratic in form. The apparently atypical results observed in Koch's data can therefore be reconciled with the linear trends elsewhere. It is however noted that at gap levels less than 0.24, Koch's curve is extrapolated.

5.6. Experiments with Machines of Significant Pressure Ratio

Most of the work so far reviewed has been executed either on essentially incompressible flow or moderately loaded machinery. Investigations carried out by Moore and Osborne (35) however were on tip clearance effects on a transonic four stage rig. As part of a broader investigation that examined casing treatment, experiments without casing treatment indicated a decrease of pressure ratio and efficiency with increasing tip clearance and also an effect upon the stability limit of the compressor. Fan tip clearances from .061 cm to .178 cm were employed and the fan, whose design overall pressure ratio was 1.75 was tested from 50% to 100% of its design speed. The trend of efficiency and pressure ratio change with tip gap size, while similar to that of other data was not quite linear (figure 22), the rate of loss being greater with tip gap change at small tip gaps. In a transonic machine, the profile loss always rises sharply in the outer region of the blade because of the presence of shock waves and shock/boundary layer interactions. From reference 36 it is seen that a substantial part of the rotor blade had a supersonic entry Mach number. The non-linear form of Moore and Osborne's data may then be due to the progressive removal of the radially non-linear loss associated with the shock system as well as the effects of the annulus/gap interaction.

The results of Lindsey (37) may also be cited here. His work was in the development of the Armstrong Siddeley Mamba engine where, although stage pressure ratios were signifant, yielding an overall pressure ratio on three stages tested of about 1.7:1, there was no significant supersonic flow. Tests indicated an almost linear drop in efficiency with increased tip clearance (figure 23). It is however mentioned that there was considerable scatter in Lindsey's data.

6.0. DISCUSSION OF COMPRESSOR EXPERIMENTAL DATA

6.1. Observations on the Applicability of Cascade Results and Models

The review of the literature on cascade measurements covers cascaded blade tip gap measurements for a variety of conditions:

a) With inviscid uniform flow;

b) With inviscid non-uniform flow;

c) With sidewall boundary layers of varying strength;

d) With relative movement between the blade tip and the sidewall.

It is as well to examine the manner in which these experiments simulate conditions in a real turbomachine.

The experiments of Khabbaz (4), Yokoyama (5) and some of the work of Lakshminarayana and Horlock (6, 7) were with an imaged blade in uniform inviscid flow. Although Dean (9) indicated that in such a situation none of the bound vorticity could bridge the gap to enter the annulus sidewall, this concept was countered by Lakshminarayana and Horlock (6) who introduced the idea that as the gap grew, vortex lines from the blade tip to its image were progressively cut and turned into the tip-locked trailing vortex. Their statement has already been supported by the earlier empirical argument for the inviscid flow case. The applicability of this concept to the viscous flow situation universally encountered in turbomachinery must however be investigated

Appealing to Helmholtz's second and third laws of vortex motion which are applicable in an inviscid, incompressible flow, a vortex filament must form a closed loop or terminate at a boundary. In the instance of a finite wing in an infinite flow field, the vortex filament bound to the wing forms a closed loop with the downstream convected starting vortex by the shed wing-tip vortices. For a two-dimensional cascaded aerofoil in an inviscid uniform flow field there is a similar vortex loop, the shed vortices being contained conceptually within the sidewalls. In the case of the experiments described above in which a blade was terminated at one end at the sidewall and at the other end by a tip gap which reached to an image blade, and assuming that the sidewall flow was inviscid there was then a closed loop of vorticity locked to each aerofoil individually, one streamwise leg conceptually within the wall and the other actually shed at the blade tip. A further loop of vorticity, part of which bridged the gap, embraced both aerofoils and was shed streamwise within the sidewalls to connect with the starting vortex. The sum of the individual aerofoil vortex loop and that bridging both aerofoils yielded, at an aerofoil, the lifting line vortex and hence the section lift.

Upon the introduction of a viscous flow to a plain cascade at its sidewalls, a new situation exists. At the wall section there is a zero velocity for the usual no-slip condition. Because of the zero flow velocity in that plane, there can be no circulation at the aerofoil section as it intercepts the wall. (In the viscous flow situation however, it is possible to have retained lift at the sidewall section as a consequence of the static pressure field generated by the blade in the moving fluid and transmitted through the boundary layer. This represents a condition of lift without circulation.) With no circulation, no vortex line can pass into the wall from the section. The shed vorticity is thus constrained to leave the blade section within the flow field but maintaining the overall picture of a vortex loop as previously described. This shed vorticity is described by Hawthorne (38) as trailing shed vorticity.

As with a solid end cascaded aerofoil, so too with an aerofoil with a tip gap separating it from an end wall; no vortex line intercepts the wall and all must be shed within the flow field even though, in the viscous flow case, there is retained lift at the wall due to the superimposed static pressure distribution.

This conclusion is however in contradiction to that of Dean (9) in considering a tip gap with the presence of a boundary layer. Because of the frictional effects present he postulated that by the vehicle of viscosity a pressure difference could be maintained, permitting the bound vorticity to cross the gap. Because of the viscosity however, the no-slip condition maintained at the wall, while admitting the pressure field present, precludes the velocity field at the wall necessary for vorticity to exist there.

The results of Lakshminarayana and Horlock (6, 7) were interpreted, however, to indicate vorticity bridging the gap in their experiment with a centrally split blade and since, in bridging the gap, the line of symmetry of the experiment was crossed, vorticity conceptually entered the pseudo-wall represented by the line of symmetry. Such an experiment is not then representative of the physics of a compressor blade with clearance in a casing through which a real fluid is passing. The only way in which the correct vortex shedding condition could be obtained at the gap would be if, at a plane orthogonal to the blade section, the streamwise flow was zero, a condition that can exist only if a splitter plate is fitted. This though introduces a boundary layer at the gap.

The cascade geometry used by Lakshminarayana and Horlock in which a splitter plate was fitted, transporting a boundary layer within the gap was more realistic and in fact would be representative of a cantilevered and shrouded stator assembly or inlet guide vane row in a compressor.

The effect of blade/wall relative motion was not covered in references 4, 5, 6 or 7, but this was investigated by Gearhart (11) and Dean (9) and although their results have certain qualitative differences possibly due to geometric variations; the convection effect due to the wall movement, though not as powerful as the effect of the presence of the gap alone, was seen to be important.

No cascade experiment is however able to investigate radial effects, centrifugal force on the blade boundary layer and th consequence of radial variations in the blade flow due to design considerations, so it remains to examine data from real turbomachines to evaluate these.

6.2. Observations on the Experiments with Compressors

The data which have been published on tip gap effects measured experimentally on compressors covers nearly fifty years and, considering the duration of investigation, is rather sparse. In view of the developments in instrumentation and techniques over that period and of the profound improvements in turbomachines, the general consistency of the data are nevertheless the more remarkable.

Few measurements exist of the radial variations of the flow within the annulus, but of those that do, Hutton (20) shows an effect of tip clearance that dominates the tip region aerodynamics and has a marked effect over most of the annulus height. This indicates that beyond installing a simple loss statement in the calculations for the tip region or even allowing for recambering of the blade in the wall boundary-layer region as suggested by Daly (26) in designing compressors to allow for tip gap effects, consideration should be given to the effect over the whole of the flow field. Williams (19) measured data which while showing a marked effect of tip clearance found that it was mainly confined to the outer 25% of the blade height.

The most widely available data consists of the measurement of loss in overall pressure rise, work coefficient and efficiency with increase in tip gap. Since with increase in tip gap the reduction in swept area of the blade is $d\lambda\pi D$, linear with $d\lambda$, the change in tip gap and the work derives from the change in whirl velocity across a rotor row, a linear fall in work coefficient is reasonable. A linear change in work coefficient with an associated linear change in efficiency would not produce a linear change in pressure rise as has been indicated in assessing Koch's (34) data. Bowden and Jefferson (32), Lindsey (37), Hutton (20), Moore and Osborne (35) and Smith (33) all indicate a linear response of pressure coefficient, while Jefferson and Turner (20), Ruden (14) and Rains (17) discovered a closely linear response of work coefficient to tip gap. It therefore remains to be seen which, if either of these parameters is, in general, linear in response.

A further point of disagreement between the data examined is concerned with the efficiency variation. The measurement of efficiency always carries problems, but while absolute levels may be difficult to guarantee, relative changes on one machine in which the tip gap is being changed should maintain good internal consistency. The general linearity of response of efficiency with tip gap is confirmed by most researchers, Hutton (20) being the exception. At very small tip gaps however, a peak in efficiency was noted by Spencer (21) and confirmed by others (24, 25, 26, 27) and a similar trend was noted by Lindsey (37) suggesting, as Lakshminarayana and Horlock (6, 7), that an optimum gap exists. The work of Ruden (14), Rains (17), Williams (19), Bowden and Jefferson (32) and Koch (34) indicates no optimum or progressive rise in efficiency being measured with reducing tip clearance.

Whether an optimum gap exists in a reasonable engineering range of size remains to be seen.

Stability limit is always of importance to the turbomachinist and the data indicates that this can be affected by the tip clearance. Most workers who measured the stability boundary of their machines found that an increase in tip clearance caused the stability limit to be encountered at higher mass flow and lower pressure rise. These included Ruden (14), Hutton (20), Smith (33), and Moore and Osborne (35), whose data indicated that the greatest change took place while the gap was still small. Bowden and Jefferson (32) produced a result not repeated elsewhere, indicating an initial reduction in the stall mass flow before a later modest increase with tip gap size. Spencer (25) using a water pump indicated that the cavitation line was moved to lower mass flows with increasing tip gap, suggesting that the very low pressures associated with tight vortices shed at the tip were relieved. Jefferson and Turner (20) using various shrouded geometries showed that with increasing shroud gap the stall line was forced consistently to lower mass flows, but that an increased stator gap (unshrouded) moved the line to higher mass flows.

While the general consensus is that an increased tip gap affects the stability limit margin deleteriously Jefferson and Turner's data, which must be regarded as atypical because of the quite different geometry, nevertheless indicates that the effect is not a consequence of mere leakage flow past the tip, but must be because of complex aerodynamic interactions as yet to be understood.

Within the research area of unsteady flows in turbomachines rotating stall has long been of interest, but little is yet known of the detailed mechanics of the stall cell. It is reasonable to assume that for a tip located stall cell, the mechanics may be affected by tip gap flows and it is significant that Hutton (22) found, not only a change in the position of the stability limit line but also in its characteristic, a strong indication of a change in stall cell geometry and mechanics.

7. PREDICTIVE METHODS FOR USE IN COMPRESSOR DESIGN AND ANALYSIS

Of the various approaches to predicting the loss in compressor efficiency due to the presence of the tip gap that due to Lakshminarayana (31) is based upon earlier cascade measurements (6, 7) and that due to Smith (33) is based upon measurements made with a multi-stage low speed compressor.

Lakshminarayana (31) derived a semi-theoretical expression for predicting the decrease in efficiency due to clearance. His model was based upon use of the fraction of aerofoil lift retained at the blade tip which he approximated by the equation:

$$(I - K) = 0.23 + 7.45(\lambda/S)$$

in the range $0.01 < \lambda/S < 0.10$. Using an expression for the induced drag in inviscid flow which included the retained lift term, the lift coefficient and geometric details both of the blade and the gap, he derived an expression for stagnation pressure loss due to the potential vortex. To this he added a term to account for the kinetic energy associated with the spanwise flow in the blade boundary layer. Dividing the sum of the total pressure loss by the isentropic pressure rise in the machine he found that

$$\Delta\eta = \frac{2\Delta P}{\rho U^2 \psi}$$

Making simplifying assumptions for the displacement thickness of the boundary layers on both suction and pressure surface there resulted the semi-empirical relationship

$$\Delta\eta = \frac{0.7\lambda\psi}{Z \cos \alpha_m}\left[1 + 10\sqrt{\frac{\phi}{\psi} \cdot \frac{\lambda A}{Z \cos \alpha_m}}\right]$$

Lakshminarayana found good agreement in predicting the results of Jefferson and Turner (20) derived from Bowden and Jefferson (32), and with the data of Williams (19), Ruden (14), and Spencer (21). The model diverged somewhat from the data of Koleshnikov (39).

In an attempt to predict the flow field in the tip region of a blade Lakshminarayana introduced a theoretical model which took into account the presence of a vortex core. Within a core radius predicted by Rains (17), Lakshminarayana assumed forced vortex flow and outside of this radius, free vortex flow. The location was determined by knowing the flow field due to image vortices (as in his experiment) and assuming that the vortex originated at the leading edge, an observation of Rains (17). Extending Lamb's (40) solution for the induced flow field of an infinite row of point vortices to two infinite rows, it became possible to predict the velocity components in the field and the air angles. Lakshminarayana acknowledged that the model based upon inviscid assumptions was unable to predict the flow losses and that, according to Newman (41), the major departure between an inviscid and viscous model, so far as rotational velocities were concerned, was confined to the region at the outer edge of the core. Introducing Newman's (41) equation and modifying it by use of Batchelor's (42) solution for the deficit in longitudinal velocity, Lakshminarayana produced an average stagnation pressure loss equation of the form:

$$\frac{P_1 - P}{\frac{1}{2}\rho V_1^{\,2}} = \int_0^1 \frac{(P_1 - P)_{y/s}}{\frac{1}{2}\rho V_1^{\,2}} \cdot d\left(\frac{y}{s}\right)$$

The model predicted the flow picture from earlier cascade results (7) qualitatively with good agreement in the spanwise distribution of the passage averaged air leaving angle.

Using the observation of Lakshminarayana and Horlock (7) that the strength of the secondary flow was proportional to the shed vortex strength

$$\Gamma = (1 - K)\,\Gamma_{2D}$$

Adkins and Smith (43), assuming that the induced pitchwise-average cross-passage secondary-flow angle distribution could be obtained by taking the vortex core to have vorticity uniform in the pitchwise direction and varying as a sine-wave first half cycle in the spanwise direction, modelled the tip clearance secondary flow in a design method. They took the spanwise influence of the vorticity to be 6.5 times the tip clearance and they also allowed for some loss in the cascade pitchwise average turning by reducing the primary flow turning parameter ($\tan \alpha_1 - \tan \alpha_2$) by an amount related to the shed vortex strength. This amount was determined as ($\alpha_2{}^* - \alpha_1$) and was gained from the equality:

$$(\tan \alpha_1 - \tan \alpha_2{}^*) = K^*(\tan \alpha_1 - \tan \alpha_2)$$

where $(1 - K^*) = \tfrac{1}{4}(1 - K)$ with the factor $\tfrac{1}{4}$ determined empirically.

Smith (33) used his measurements, which have already been reviewed, in a technique to predict how the pressure-flow and efficiency-flow characteristics were affected by the tip gap aerodynamics. This involved a two-step calculation using, initially, cascade data to determine performance at various spanwise stations and then superimposing the effects of the hub and tip boundary layers. To do this Smith defined the displacement thicknesses of the boundary layers in the traditional manner

$$\delta_h{}^* = \frac{1}{r_h \tilde{V}_{xh}} \int_{r_h}^{r_h+\delta_h} (\tilde{V}_x - V_x)\, r\, dr$$

$$\delta_t{}^* = \frac{1}{r_t \tilde{V}_{xt}} \int_{r_t-\delta_t}^{r_t} (\tilde{V}_x - V_x)\, r\, dr$$

and introduced a tangential force loss in the endwall boundary layers in a similar format:

$$\nu_h = \frac{1}{r_h \tilde{F}_{yh}} \int_{r_h}^{r_h + \Delta_h} (\tilde{F}_y - F_y) r dr$$

$$\nu_t = \frac{1}{r_t \tilde{F}_{yt}} \int_{V_t - \Delta_t}^{V_t} (\tilde{F}_y - F_y) r dr$$

where ν_h and ν_t represented the amount that the tangential component of the blade force was reduced from its free-stream value by the presence of the boundary layer.

For high hub/tip ratio stages this yielded the approximate relationship for efficiency:

$$\eta = \tilde{\eta} \frac{1 - \frac{\delta_h^* + \delta_t^*}{h}}{1 - \frac{\nu_h + \nu_t}{h}}$$

Smoothed data relating the hub and tip boundary layer displacement thicknesses δ_h and δ_t, and the hub and tip tangential force thicknesses ν_h and ν_t to the pressure rise as a fraction of its maximum value and the tip gap size were necessary. With respect to the assessment of boundary layer thickness, reasonable curves could be drawn through the data, but the data relating to the tangential force thickness had a high level of scatter, from which he chose a constant value of 0.65 of the displacement thickness as representative. Smith was able to conclude however that since the value of tangential force thickness was positive, stage efficiency would be automatically reduced and would, for example, be less than that predicted by Mellor and Strong (44).

The mass flow coefficient, allowing for the presence of the boundary layers was assessed in the knowledge of the displacement thicknesses of the hub and tip boundary layers, thus:

$$\phi = \tilde{\phi} \left| 1 - \left(\frac{\delta_t^*}{g_t} + \frac{\delta_h^*}{g_h} \right) \overline{g/h} \right|$$

A somewhat simplified form of Smith's model was used in a method of Koch and Smith (45) to establish the design point efficiency of a multi-stage compressor. They lumped the hub and tip boundary layer data, both displacement thickness (figure 24) and tangential force thickness (figure 25) together, producing a relationship between the non-dimensionalised sum of the two displacement thicknesses and the non-dimensionalised pressure rise coefficient thus:

$$\frac{2\overline{\delta^*}}{\overline{g}} = \left| \frac{2\overline{\delta^*}}{\overline{g}} \right|_{\overline{e/g} = 0} + 2\frac{e}{g} \left| \left\{ \frac{\Delta p_R + p_S}{q_{1_R} + q_{1_S}} \right\} \middle/ \left\{ \frac{p_R + p_S}{q_{1_R} + q_{1_S}} \right\}_{max} \right|$$

As with the individual tangential force thickness data, the combined data also contained a great deal of scatter. Through this the authors put a simple line as representative. Their method also allowed for a modification in the boundary layer displacement thickness due to the axial spacing as had previously been recognised as necessary.

Koch and Smith were able to conclude that the end wall boundary layer, represented by its displacement thickness yielded an efficiency decrease, but this was partly offset by the deficit in the tangential blade force in the boundary layer.

8. TURBINE TIP GAP DATA

Although there are obvious geometric similarities between compressor and turbine rotor assemblies in the region of the tip gap, two dominant features are likely to create different aerodynamic effects.

a. Because the flow across a turbine blade is accelerating,

   i) The wall boundary layers are not so prone to separation;

   ii) The pressure distribution around the blade profile is rather different to that of a compressor. With a compressor blade profile the maximum pressure difference across the tip occurs towards the leading edge and it is this that probably locates the tip shed vortex observed and used in certain of the models. A turbine blade profile however in general maintains a more constant chordwise pressure difference across the tip and it may be anticipated that this will alter the geometry of the flow.

b. The turbine rotor/wall relative speed is in the opposite direction to a compressor rotor. The immediate consequence of this is that the tip leakage flow which is from the suction to the pressure surface and is now additive to the channel generated secondary distributed circulation.

It may then be anticipated that the tip gap behaviour of a turbine would be rather different to that of a compressor.

9. TURBINE CASCADE EXPERIMENTS

Using a water-table and a series of simple geometries Booth, Dodge and Hepworth (46) examined the flow across a tip gap without relative wall movement. They identified three regions in which different terms of the momentum equation were dominant.

1) At low clearance levels, where shear forces directly balanced pressure forces;
2) At normal clearance levels where convection became dominant;
3) At high clearance levels where pressure gradients became small because of tip unloading.

Their series of tests showed the flow to be primarily inviscid with a high discharge coefficient insensitive to flow on the blade suction surface.

In further tests in which various geometric alterations were made to the tip section, a reduced thickness tip section produced a noticeable performance degradation and while at a particular thickness the degradation reduced with reduced tip gap the effect of the tip thickness became greater.

Wadia and Booth (47) noted that, for a turbine, discharge coefficients were reduced due to wall motion because the relative movement of the wall was against that preferred by the leakage flow (in a compressor the opposite argument holds).

Wadia and Booth also presented computed streamline patterns in the tip region showing circulation above the suction surface adjacent to the tip region. While these data are rather similar to measurements on compressor cascades, e.g. Dean (9) and Peacock (10), they do not coincide with turbine cascade data of Peacock (48) who found that the vortex migrated an appreciable distance over the blade span.

10.0. MEASUREMENTS WITH TURBINES

10.1. Effect upon Efficiency of Tip Gap

The centre of interest in tip gap effects in turbines is with the small diameter machine of which the tip gap is always comparatively large related to any relevant machine dimension. Its effect upon a small machine is in consequence larger.

As a means of extending observations on the water-table to a rotating machine, Booth, Dodge and Hepworth (46) used a low aspect ratio turbine in which the tip gap was varied from 1% (.013") to 3% (.039") of the blade height. The design point total-to-total efficiency fell from 92% to 89%; 1.5% for every 1% of blade height.

Ewen, Huber and Mitchell (49) considered the blade tip clearance as one feature affecting the aerodynamic efficiency of a turbine. Progressively altering the outer diameter of the annulus, they found that in the range $.007" < \lambda < .028"$ the efficiency fell progressively, but in a non-linear manner, with increase in tip clearance (figure 26). The nature of the non-linearity, a greater loss for increments in gap dimension at small gap, was qualitatively similar to the fan data of Moore and Osborne (35). In this, the data of Ewen, Huber and Mitchell appear to be unusual, for their turbine was subsonic and most other turbine data do not show the observed effect at small gap heights.

Early work on a small impulse turbine by Kofskey (50) yielded a linear 1.75% reduction in turbine efficiency for a 1% increase in annulus height. Extending the investigation to a two-stage turbine with hgih reaction in both stages and using two tip clearances corresponding to 1.06% and 2.47% of the average annulus height, Kofskey and Nusbaum (51) also detected a linear relationship in efficiency degradation. The rate of efficiency shedding was however higher and this was attributed to the higher reaction, which would yield a larger pressure difference across the blade tip overall.

These trends were confirmed by Futral and Holeski (52) and Holeski and Futral (53) on a single stage high reaction turbine. A 20% change in static efficiency (0.79 to 0.63) was measured in increasing the rotor tip clearance from 1.2% to 8% of the blade height and the slope of the straight line was closely similar to that obtained by Kofskey and Nusbaum (51).

Szanca, Behning and Schum (54) used a 10.0" diameter turbine of 80.5% tip reaction to establish a closely linear relationship between the efficiency decrement and the rotor tip clearance. This was varied from 2.3% to 6.7% of the annulus height by progressively machining the rotor blade tips.

Haas and Kofskey (55, 56) investigated the effect of the rotor tip on a 5.0" turbine, but varied the geometry to include both a recessed wall with extended rotor blade as well as a plain outer wall and normal tip geometry. While the rate of which efficiency loss with increase in tip gap was lower with the recessed wall geometry, in both cases, straight line relationships resulted. Since, in the case of the recessed wall geometry, the blade tip gap was always within the recess and therefore not subject to free-stream or even normal wall boundary-layer conditions, but may have been in a quasi-stationary region within a flow separation created by the recess, it may be anticipated that the tip gap effect was somewhat suppressed.

Haas and Kofskey's papers (55, 56) are particularly valuable in that they produce a synoptic graph showing data from a range of workers (figure 27). From this, a general conclusion can be drawn, that the greater the reaction, leading to increased pressure difference at the blade tip, the greater the penalty to efficiency of increased tip clearance.

10.2. Effect upon Mass Flow of Tip Gap

A turbine rotor represents a resistance to the flow of a working fluid in an annulus and it may be anticipated that increasing the tip gap would reduce this resistance, permitting under similar inlet conditions an increased mass flow.

This was measured by various workers and, in particular, Kofskey and Nusbaum (51) and Holeski and Futral (53) (figure 28) found that the relationship was linear. Such a relationship might be anticipated since the variation of the tip gap area is linear with tip gap height as long as the tip gap is small compared with the annulus outer diameter.

Since the increased mass flow is likely to be passing through the gap region rather than the turbine rotor channel, this would not affect the turbine work coefficient which would change directly with the measured efficiency.

10.3. Spanwise Effects of Tip Clearance

Detailed measurements by a number of workers have indicated that the effects of changing the tip gap, while profound in the tip region, are not confined to it. Ewen, Huber and Mitchell (49) measuring the efficiency at different radial stations, found that, in the tip region, up to 15% loss of efficiency could be measured in quadrupling the tip gap (figure 29). While it reduced towards the hub, the decrement was still measurable at 40% of the blade height. Kofsky and Nusbaum (51) measured the effect upon the exit stagnation pressure of increasing the gap 2.40 times to extend about 14% of the blade height from the tip and over this region the exit angle also increased, thus contributing to the loss of efficiency. Holeski and Futral (53) and Szanca, Behning and Schum (54), however, measured an increased flow angle with tip gap over the whole of the blade height (figure 30a), while Holeski and Futral's data indicated that the change in stagnation pressure remained fairly localised in the outer 20% of the blade height (figure 30b). Contrary to the data of Ewen, Huber and Mitchell (49), Holeski and Futral also measured an efficiency decrement over the whole blade height (figure 30c).

It is clear from these data that the effect of the tip gap aerodynamics upon the aerodynamics of the whole blade is very large and it may be reasoned that design procedures used over the whole blade height should include an allowance for the tip flow.

11. SOME ENGINEERING CONSIDERATIONS

Most of the work considered in this review has been executed under somewhat idealised conditions where the tip gap had a well defined constant value in any particular test. Because of inherent engineering features the tip gap of a turbomachine may not always be constant.

Blade growth due to rotational speed and temperature change automatically reduces the tip gap and this was recognised in early investigations by Lindsey (32), who was forced to relate all of his data to a nominal cold clearance of 0.040". Ewen, Huber and Mitchell (49) were able to quantify the effects of centrifugal growth of a turbine in which the tip gap reduced from .014" to .010", contributing about 25% of the increase of the 4% efficiency change in changing the operation from a velocity ratio of 0.65 to 0.75. While such an observation diminishes slightly the effect of tip clearances upon turbomachinery operating at design it must also be borne in mind that at reduced speed the tip gap would be increased from such a desirable minimum value and a performance penalty would be incurred, in consequence.

An attempt to meet the problem of the change in tip gap due to temperature and rotational effects during an engine cycle has been made by Beitler, Saunders and Wagner (57), who by selective cooling of the turbomachinery casings controlled the tip gaps.

Ovality in turbomachinery casings has traditionally presented problems to the engineer. Simulating a range of in-flight loads upon an operational aircraft engine Stakolich and Stromberg (58) measured the changes in tip clearance at all compressor and turbine stages through the machine. They concluded that over a period of 2,000 flight cycles a 2% change in thrust would occur, mainly due to thermal distortion and erosion of the tips.

An oval casing to a turbomachine produces a time-wise change i the tip gap at any particular blade, but a further problem of non-uniformity is that of blade-to-blade variations in tip clearance.

This is an area which has not been specifically addressed, but there is indirect evidence that such variations have an effect upon performance of a turbomachine. Ramachandra (59), investigating a reported noise problem in a particular aircraft cockpit, found a high noise content at 166 Hz. At the compressor operational speed of 10,000 rev/min this represented the passing frequency of one blade in a blade row. Now, if the noise was generated due to casing ovality, the dominant frequency would be

$$f_N = \frac{N}{60} \cdot n,$$

where n is the number of blades, but since it was at the frequency

$$f_N = \frac{N}{60},$$

the blade passing frequency, it may be presumed that in the particular case Ramachandra was examining, there was a blade of non-uniform height. Since the noise was generated by some unsteady aerodynamic phenomena it may be concluded from Ramachandra's investigation that a single non-standard gap could have a more general effect upon a machine than at that blade tip and that a noticeable variation in tip aerodynamics, tip to tip, may be present.

12. FUTURE RESEARCH

Two objectives are seen in a coherent programme of research: into the effects of tip gaps upon turbomachine aerodynamics.

i) the provision to designers of a method to account in design for such effects, both in the assessment of performance loss overall and in modifying design across the blade height to allow for the resulting changes in the flow.

If we consider the tip gaps used in current turbomachinery practice it can be said generally that the penalty on compressor efficiency for the existence of a tip gap is of the order of 5% and that on turbine efficiency 4%. Regaining these efficiency losses by control of the tip aerodynamics could then yield in a gas turbine cycle an improvement in overall efficiency of the order of 5% and this would be most desirable.

To do this, however, the designer needs in his design code a sub-routine that would allow him to modify a design executed using, for example, streamline curvature and blade element design techniques, to include an allowance for the aerodynamic variations due to the tip gap. Such variations, it is seen, could extend over the whole of the blade height and must be related eventually to the tip gap geometry and local conditions.

It remains though to relate the aerodynamic variations to geometry and flow conditions at the tip so that the blade geometry may account suitably for them. Some blade designs have been proposed in which the rotor tip and hub geometry reflect by changes, for instance in local chord and camber, the inlet velocity profile. Beknev (60), Vavra (61), and Peacock (62) have considered this, but no conclusive data have yet emerged.

ii) a base set of data by which geometric modifications, such as casing or tip treatment and design changes radially along a blade, can be referred.

Various casing and tip treatment programmes, not reviewed here, have been reported and are in progress. An accurate assessment of the utility of such geometric changes will only be possible when a clear knowledge of performance with plain walls or tips can be quantified.

A coherent programme to meet the requirements (i) and (ii) identified above would include an examination of blade rows embedded within a multi-stage machine which, while it would cover the range of geometric parameters encountered in current practice, would also have as controlled variables the annulus boundary layers, the tip gap size and geometry. Measurements would include those necessary to determine overall performance parameters, radial variations of mean flow conditions and the detailed investigation of the tip aerodynamics. For such a programme instrumentation would necessarily include both that of traditional low response rate and high response rate equipment capable of discriminating the flow conditions at blade passing frequency. Of course, the accurate measurement of the tip gap at operational conditions would be essential.

13. CONCLUSIONS

For turbomachinery, both expansive and compressive, it is possible to make the general statement that the presence of a tip gap reduces performance, an effect which is progressive with increase of gap size. Because of the different relative velocities blade to wall of the compressor and the turbine the detailed mechanics of the flow leading to performance change vary somewhat. It is seen, though, that the aerodynamic effect of the tip gap is not restricted to the tip gap region but can reach over the whole of the blade height, so that adequate allowance for the tip gap in a design procedure must embrace the whole of the blade height.

Research programmes, in which certain components contributing to the overall mechanics of the flow were investigated, were seen to be inadequate in modelling the complex phenomena present as a result of the tip gap. As a consequence, techniques that would permit a process of superposition of various data to synthesise the effects of tip gaps are not anticipated to have validity.

There is disagreement between various workers in allowing for the tip vorticity shed into the working fluid and the prediction method proposed by Lakshminarayana, based upon earlier work by Lakshminarayana and Horlock, while yielding good results in the test cases cited is based upon an incorrect accountancy of the vorticity in the tip gap.

Models proposed by Smith and Koch and Smith, which are based upon measurements from blade rows embedded in turbomachines, are seen to be the most realistic, but do not yield a detailed understanding of the physics of the process and as such do not lead directly to a detailed formulation of the flow.

Data available from tests with various turbines are much more consistent than those from compressors, possibly because in an accelerating flow field, the effect of the varied boundary layers is diminished. In general the loss of performance is a function of gap size and reaction of the turbine stage, some further variation being offered in different tip geometries.

The review indicates that future research would be most profitably executed using representative geometries of turbomachine, particularly in the area of compressor research where diffusing flows heighten the sensitivity of results to the condition of the boundary layer.

A further point that emerges is that non-uniform tip gaps, due either to ovalling of the casing or to non-uniformities blade-to-blade, can have an effect upón performance. Since this represents a real engineering consideration this needs detailed investigation.

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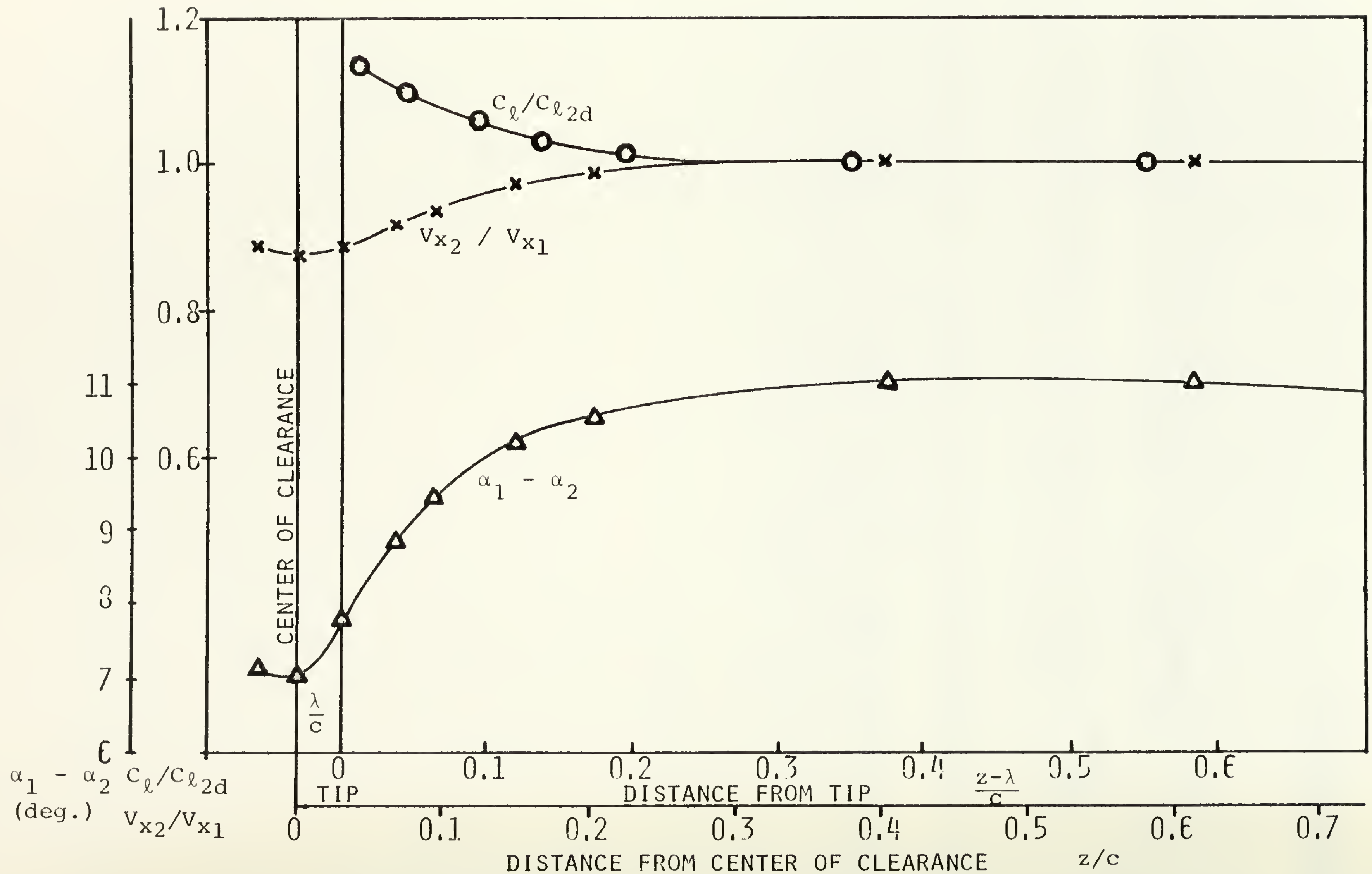


FIG. 1 DISTRIBUTIONS OF LIFT COEFFICIENT, OUTLET AXIAL VELOCITY AND TURNING ANGLE VS. SPANWISE DISTANCE (REF. 5)

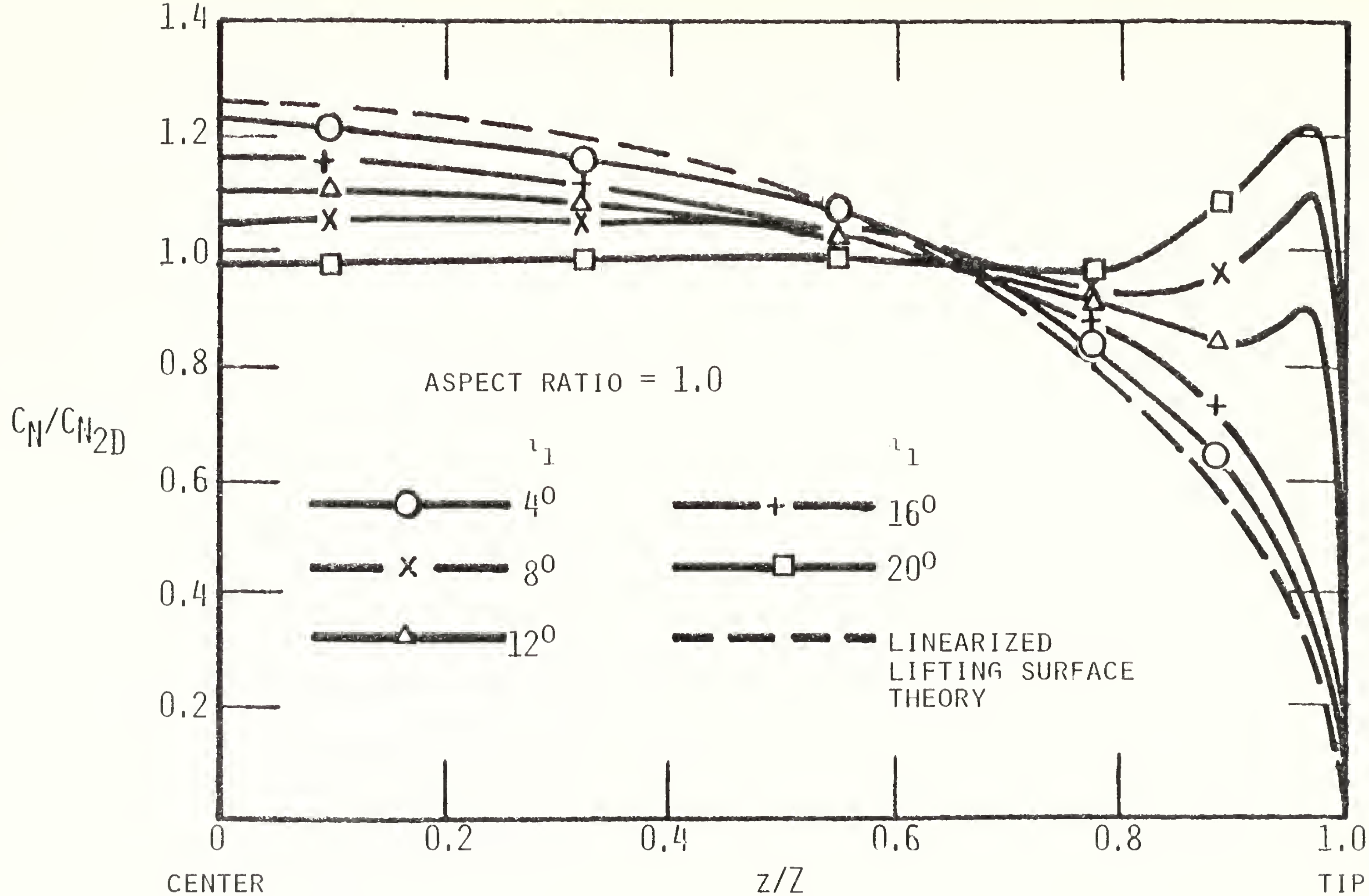


FIG. 2 NORMAL FORCE DISTRIBUTION ALONG SPAN OF RECTANGULAR WING (REF. 10)

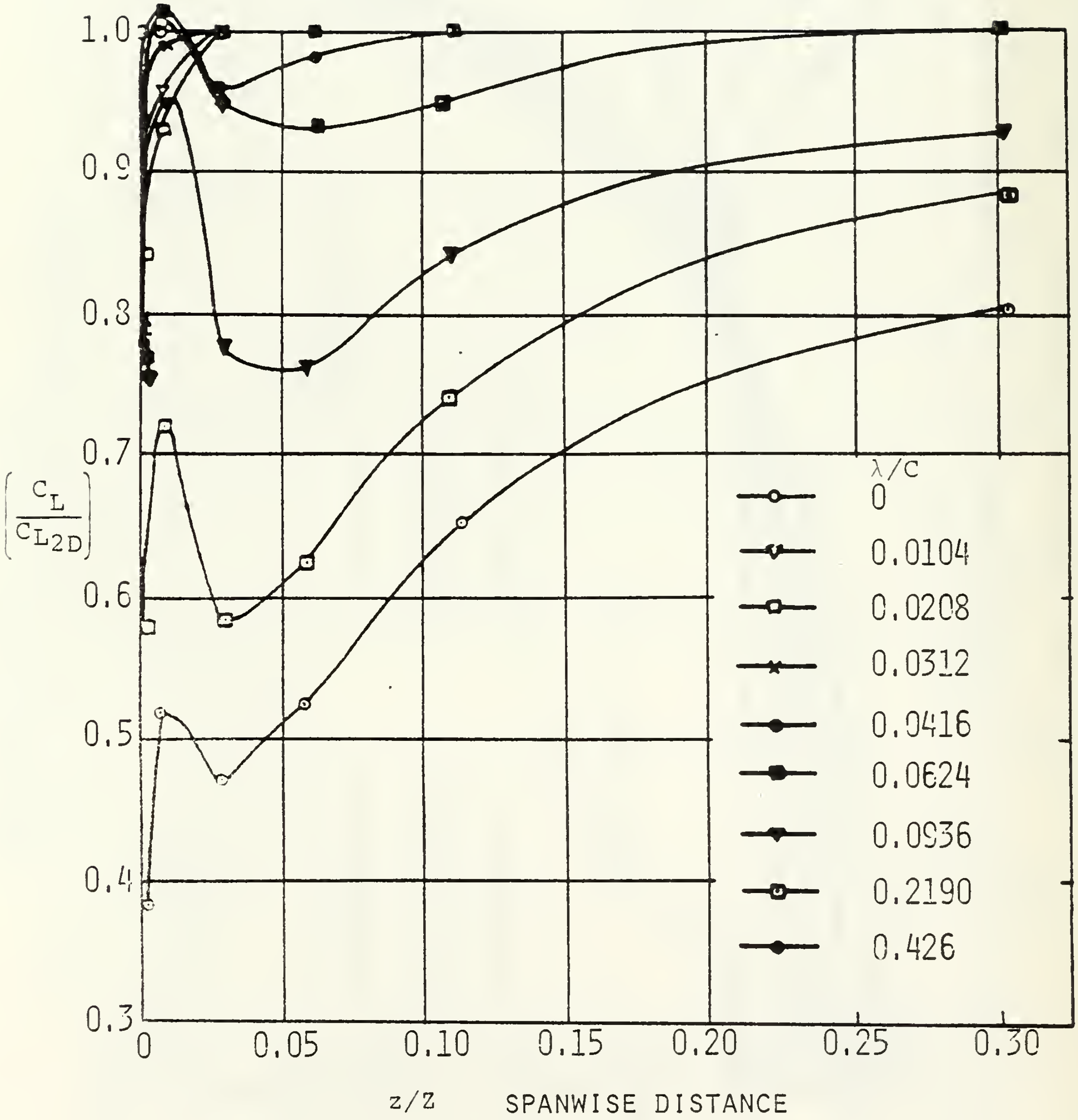


FIG 3 SPANWISE LIFT DISTRIBUTION AT VARIOUS GAP/CHORD RATIOS (REF. 6)

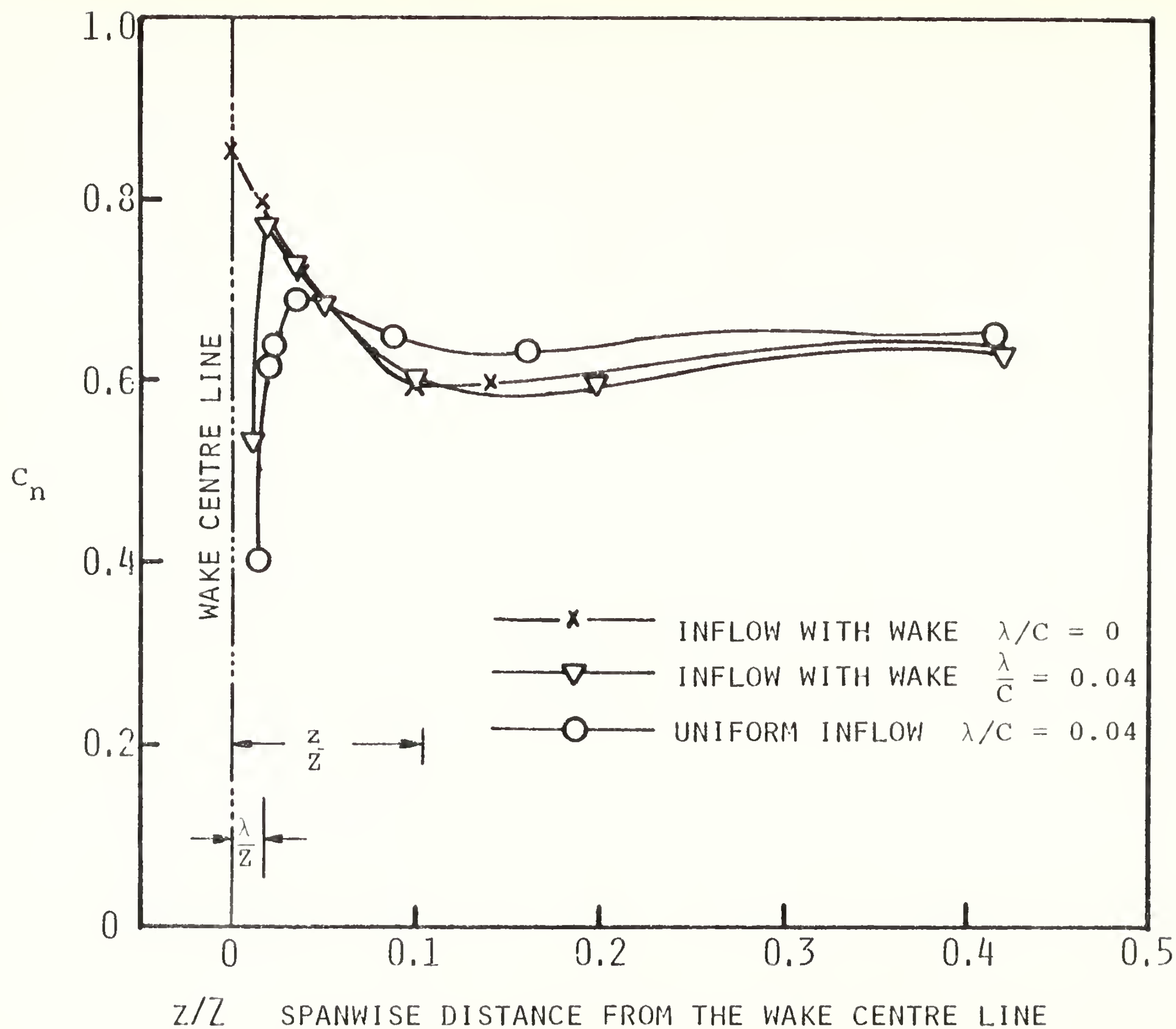


FIG. 4 SPANWISE DISTRIBUTION OF NORMAL FORCE COEFFICIENT FOR $\lambda/c = 0$ AND 0.04 (EXPERIMENT A AND B) (REF 7)

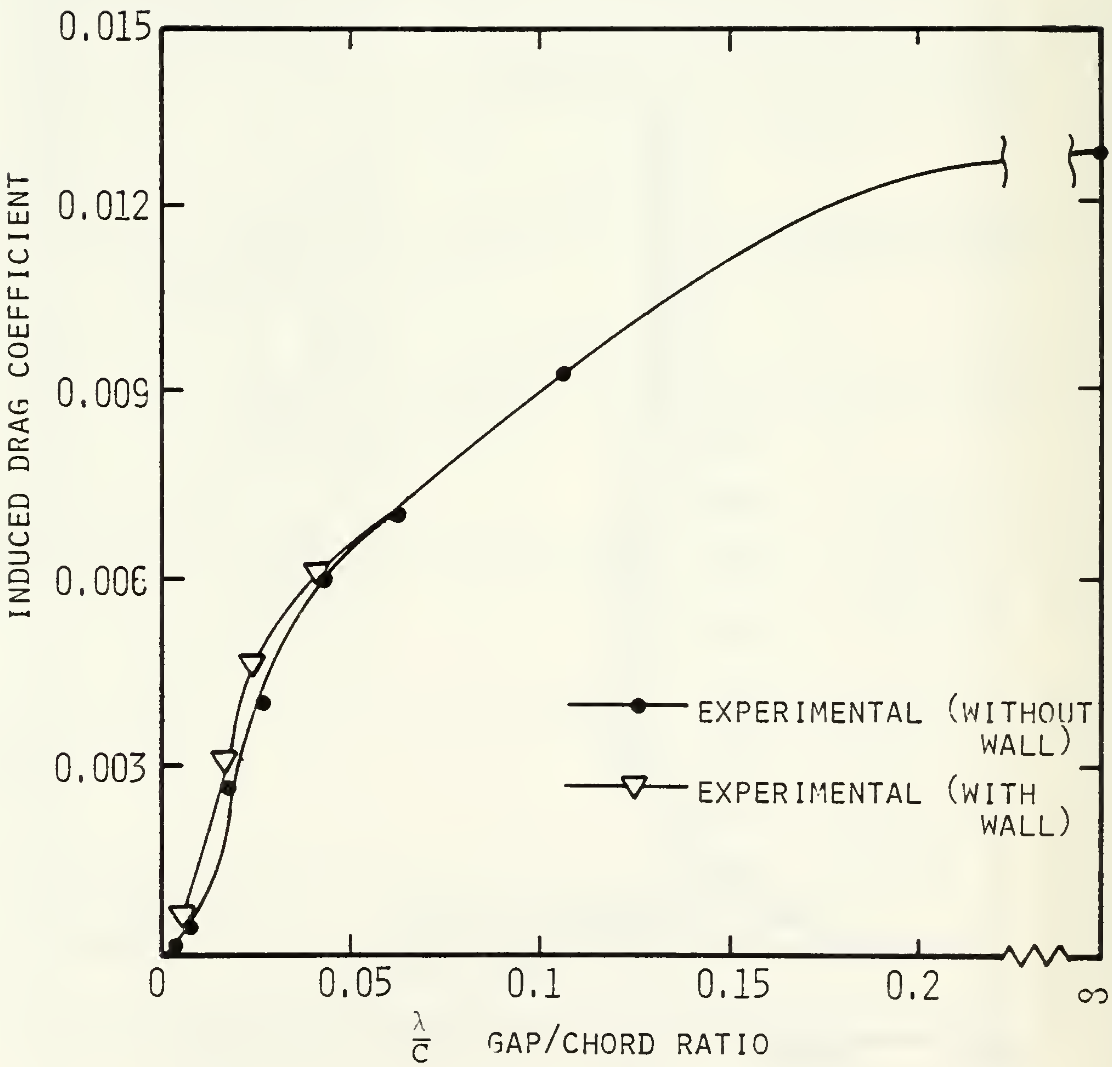


FIG. 5 VARIATION OF INDUCED-DRAG COEFFICIENT WITH λ (REF 6)

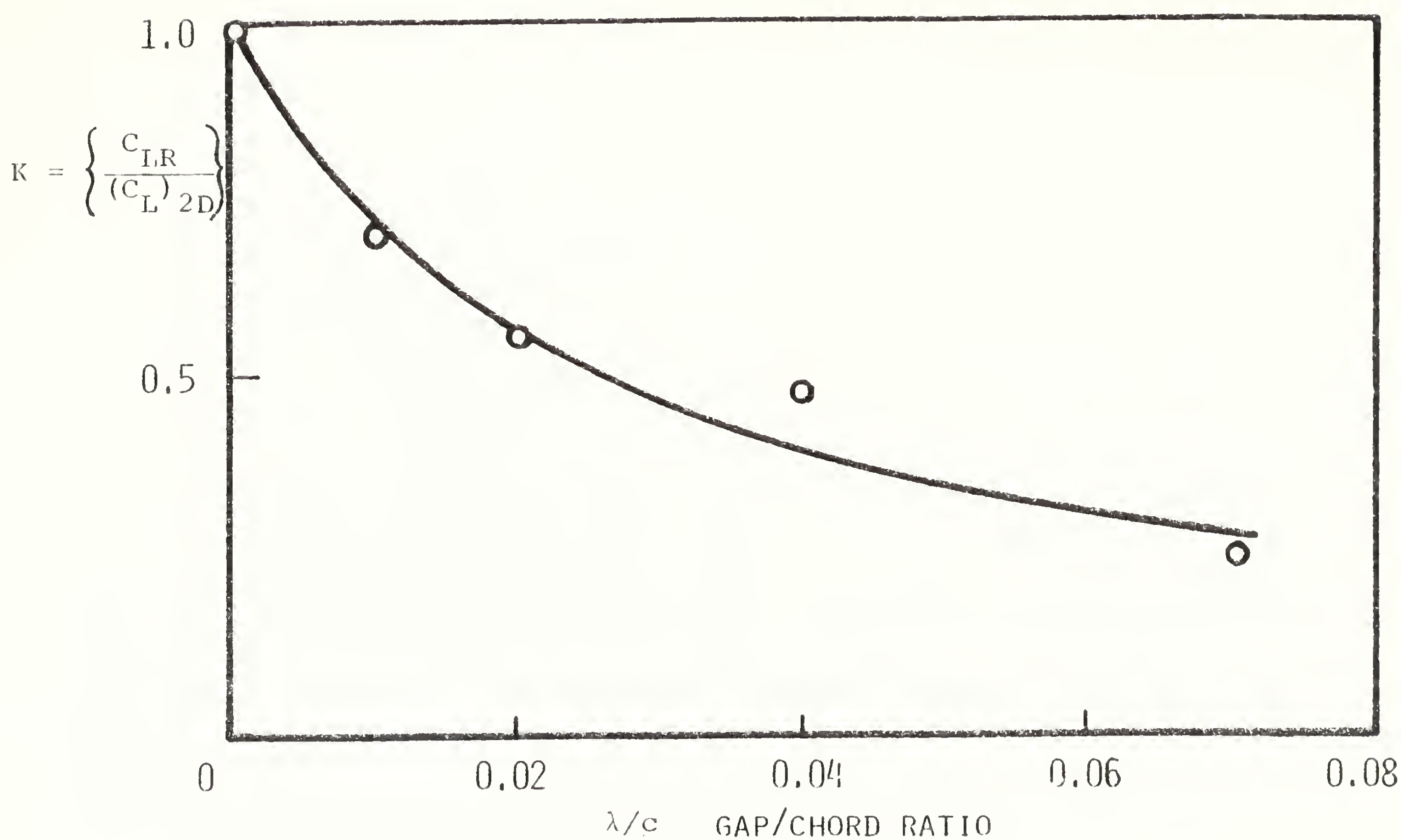


FIG. 6 VALUES OF LIFT RETAINED (AS A FRACTION OF TWO DIMENSIONAL VALUE) AT THE TIP OF THE CASCADE BLADE (REF. 7)

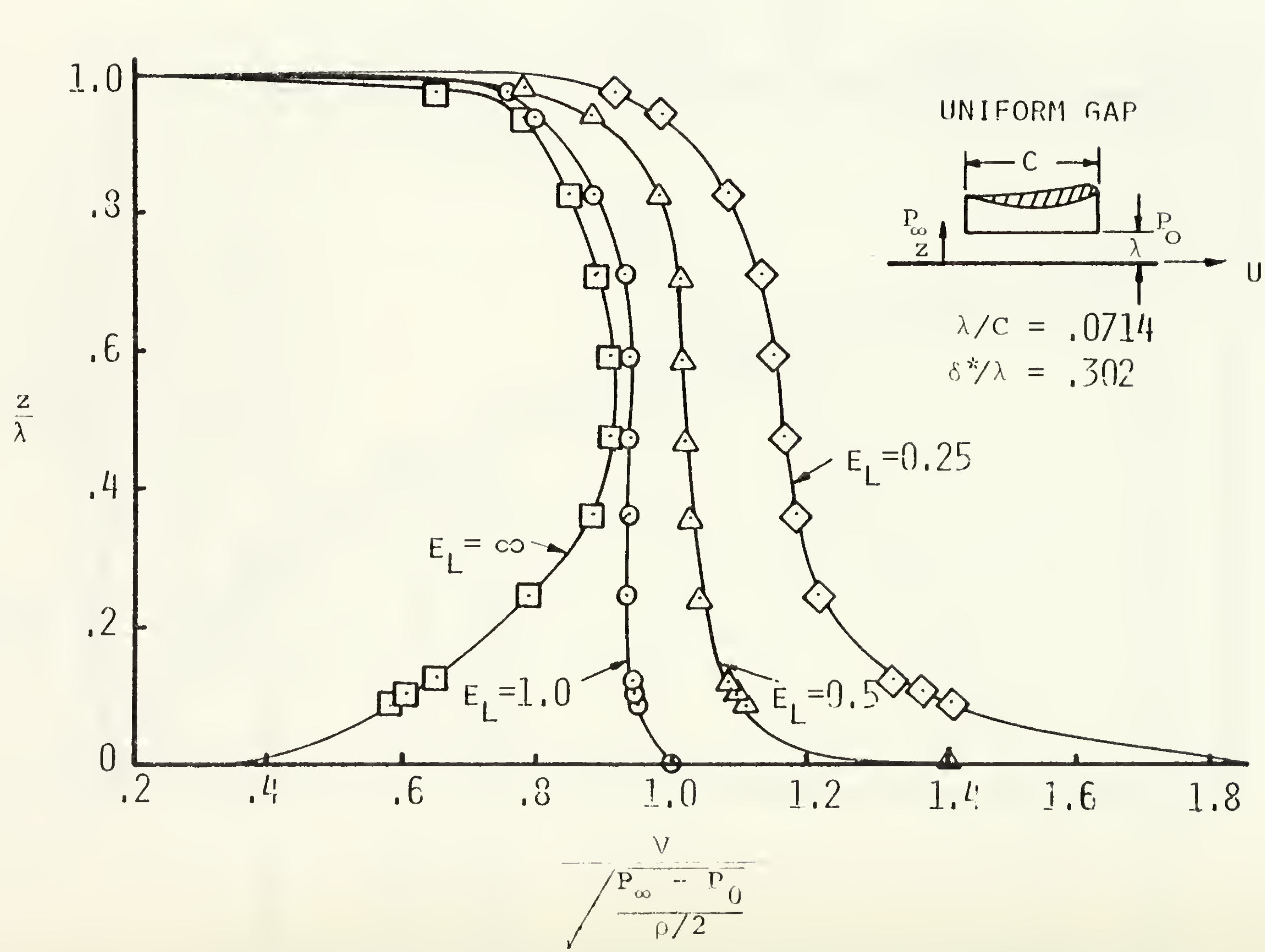


FIG. 7 NON-DIMENSIONAL VELOCITY PROFILES AT EXIT OF UNIFORM GAP (REF. 11)

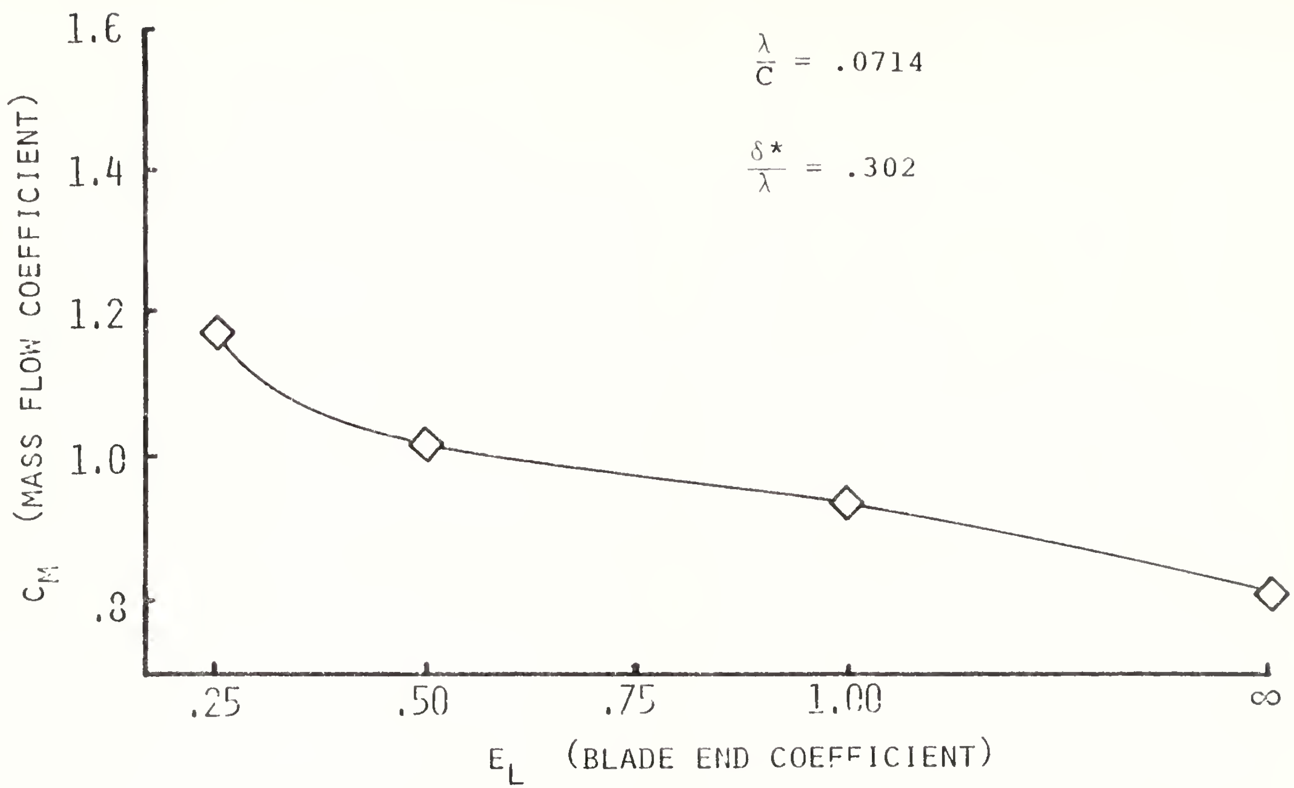


FIG. 8 MASS FLOW COEFFICIENT VERSUS BLADE END COEFFICIENT FOR VARIOUS GAP CONFIGURATIONS (REF. 11)

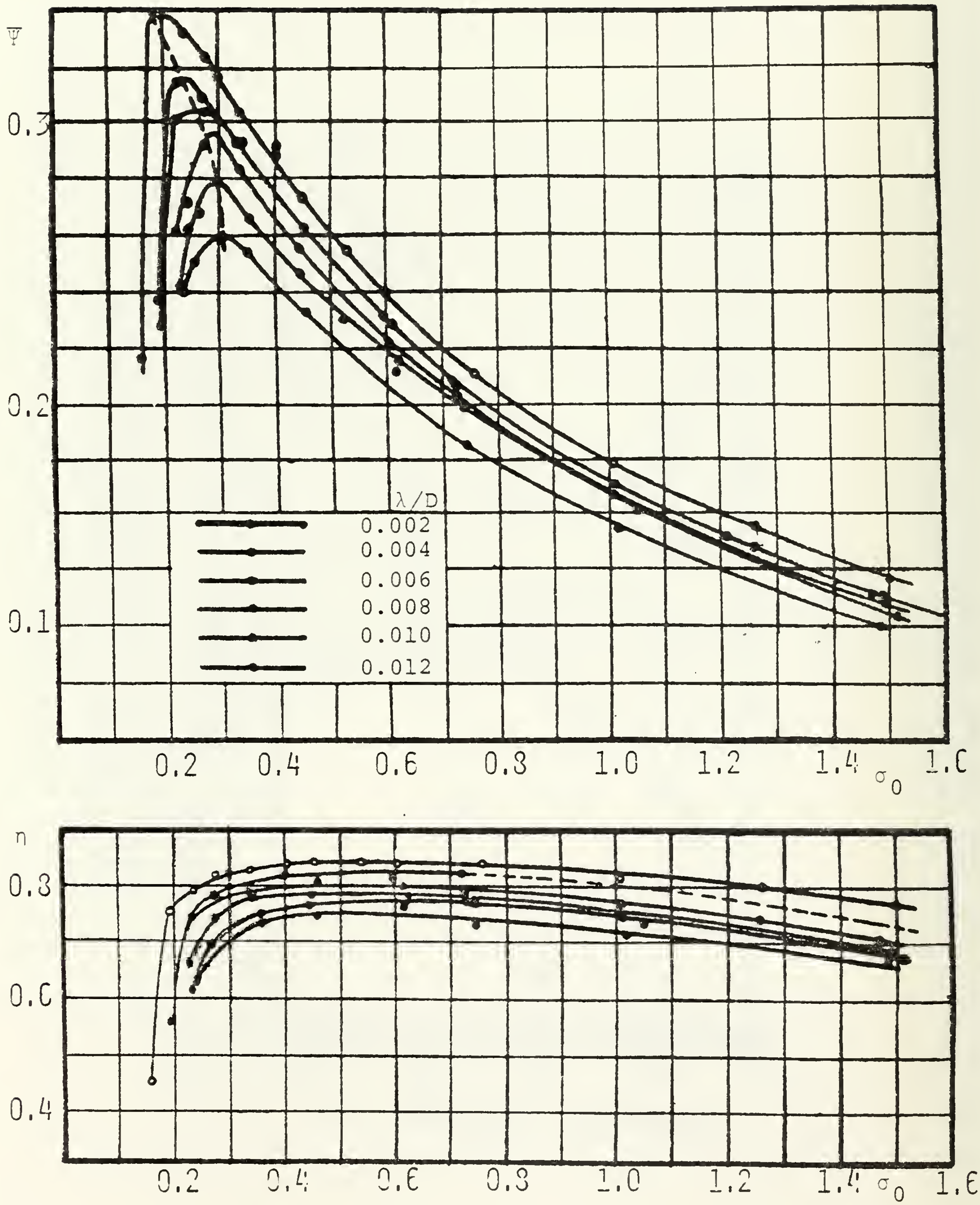


FIG. 9 EXPERIMENTAL CHARACTERISTICS FOR VARIOUS CLEARANCES BETWEEN ROTOR BLADE AND HOUSING (REF. 14)

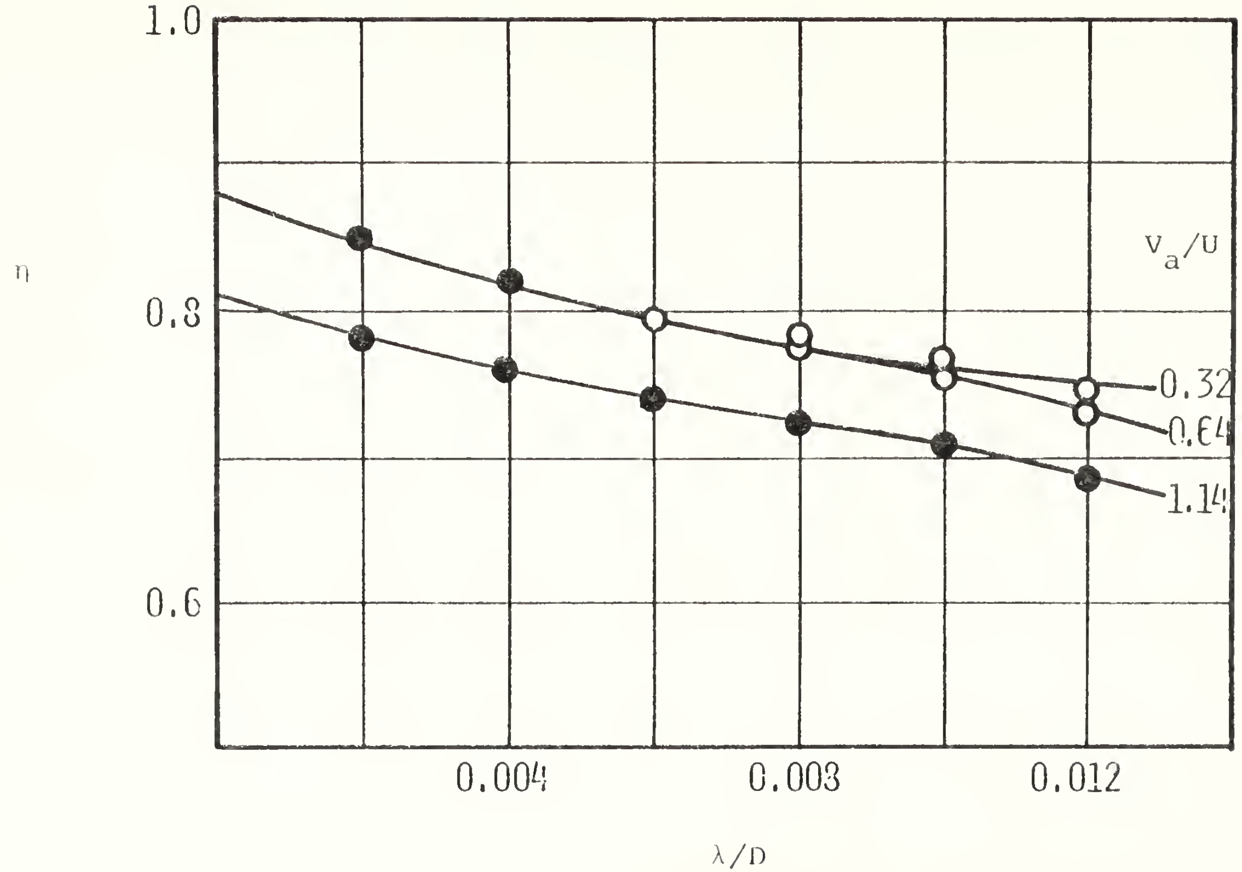


FIG. 10 EFFECT OF TIP GAP ON COMPRESSOR EFFICIENCY (REF 14)

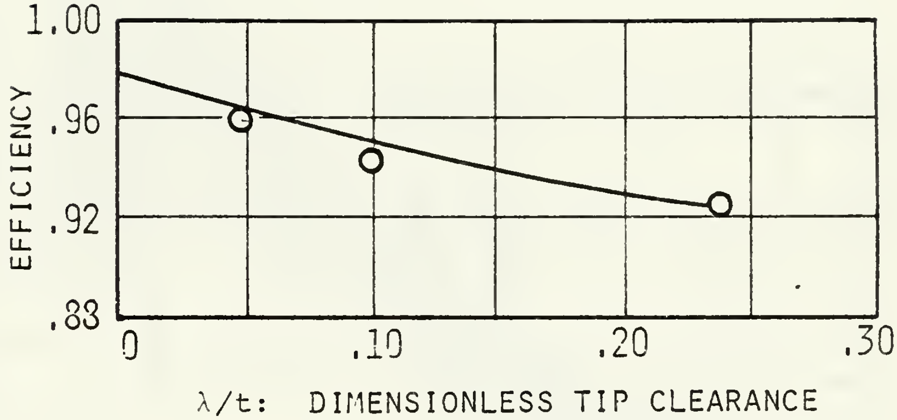


FIG. 11 VARIATION OF THE EFFICIENCY WITH ROTOR TIP CLEARANCE (REF. 17)

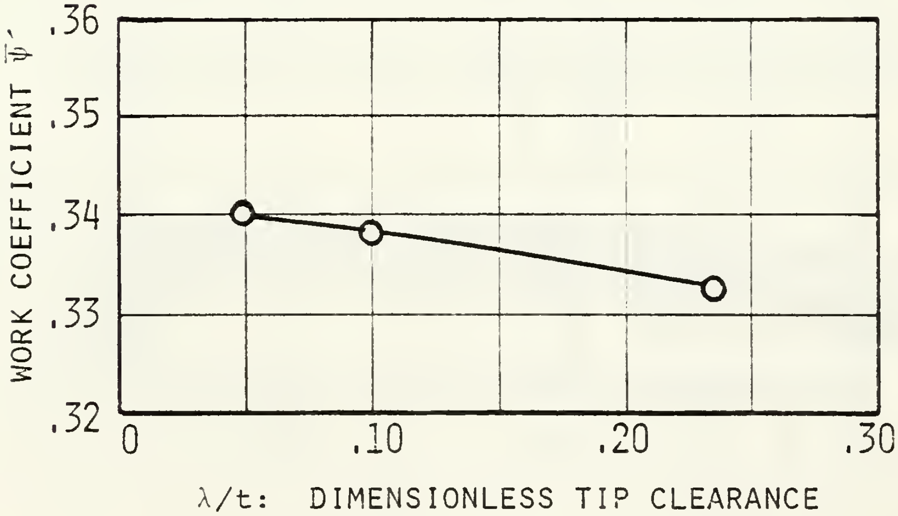


FIG. 12 VARIATION OF THE WORK COEFFICIENT WITH TIP CLEARANCE (REF. 17)

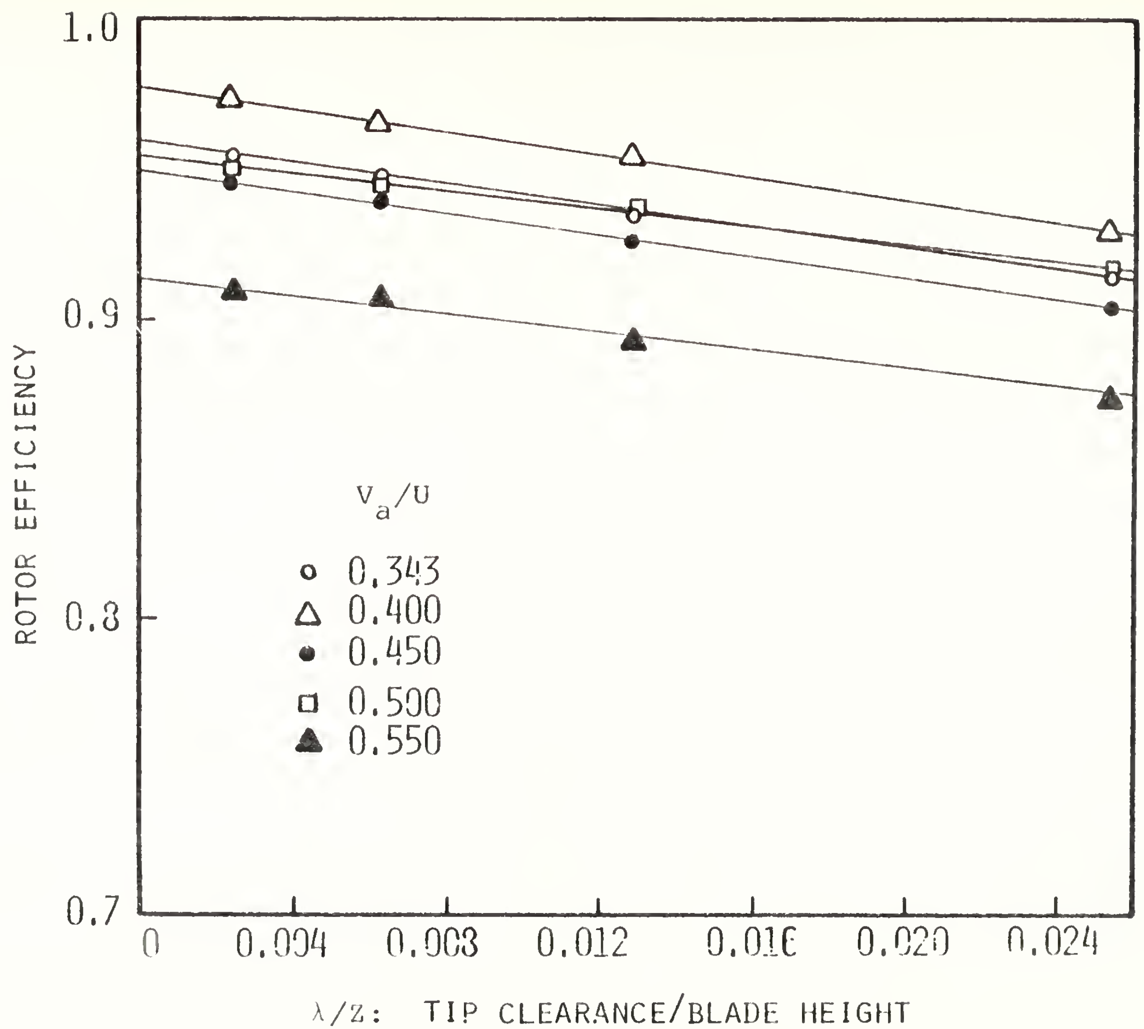
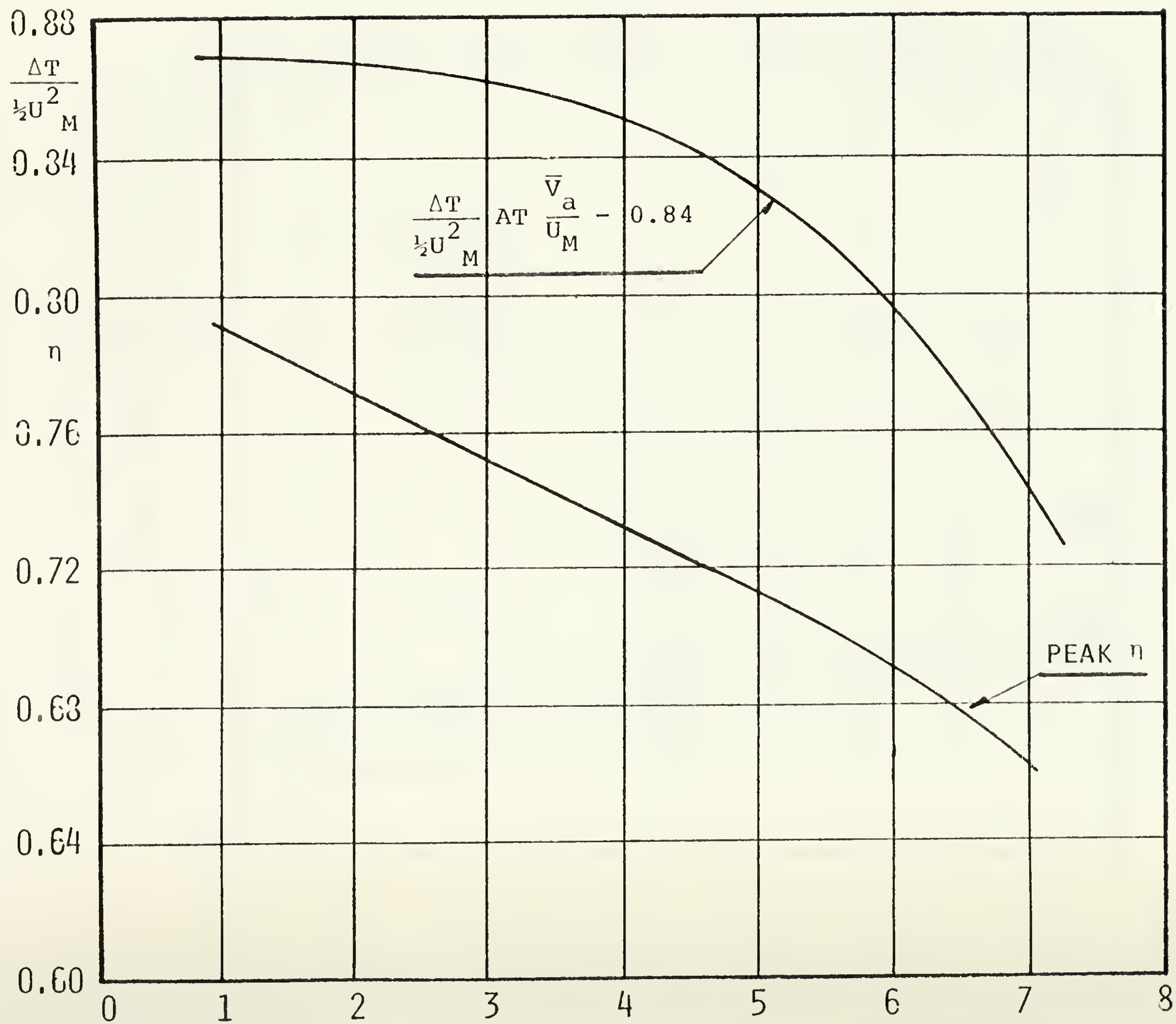


FIG. 13 ROTOR EFFICIENCY AS A FUNCTION OF TIP CLEARANCE FOR VARIOUS FLOW RATES (REF. 19)


0.88
0.84
0.80
0.76
0.72
0.68
0.64
0.60
$\frac{\Delta T}{\frac{1}{2}U_M^2}$
η
$\frac{\Delta T}{\frac{1}{2}U_M^2}$ AT $\frac{\bar{V}_a}{U_M}$ - 0.84
PEAK η
0
1
2
3
4
5
6
7
8

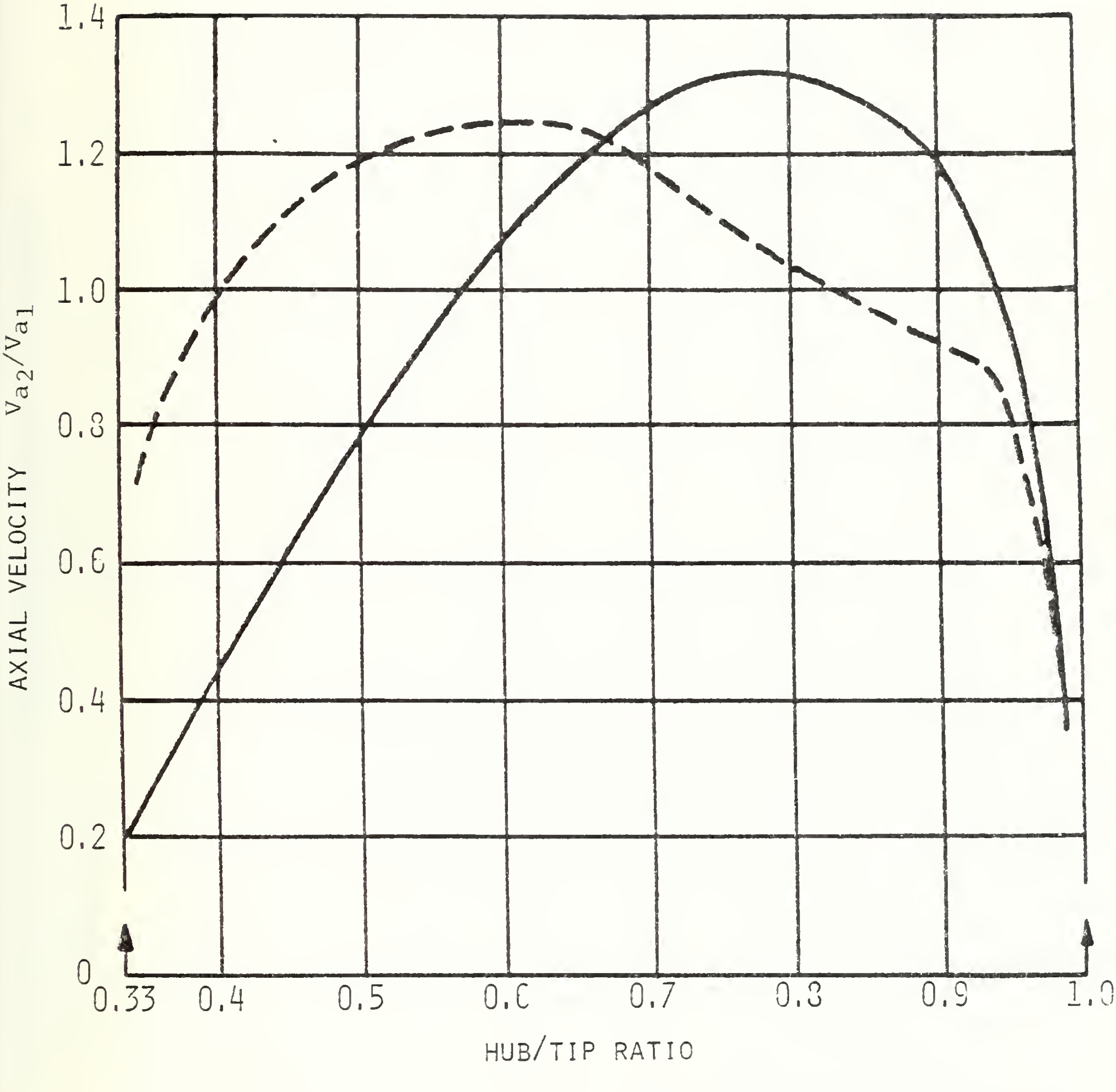
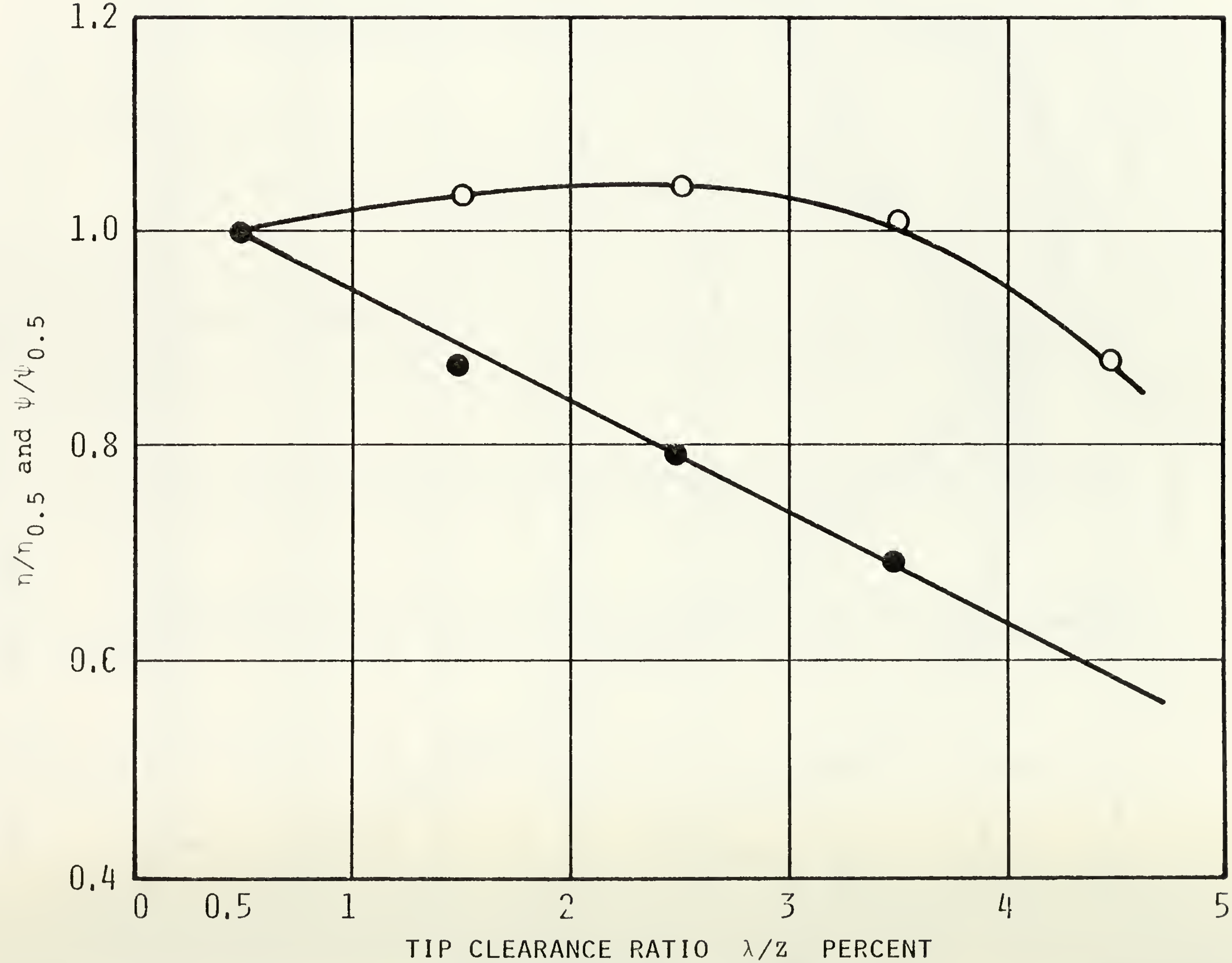


FIG. 15. THE EFFECT OF TIP CLEARANCE ON THE DISTRIBUTION OF AXIAL VELOCITY (REF. 22)

——————— SMALL CLEARANCE, λ/z = 0.5 PER CENT

— — — — LARGE CLEARANCE, λ/z = 4.5 PER CENT


1.2
1.0
0.8
0.6
0.4
0
0.5
1
2
3
4
5
$\eta/\eta_{0.5}$ and $\psi/\psi_{0.5}$
TIP CLEARANCE RATIO λ/z PERCENT

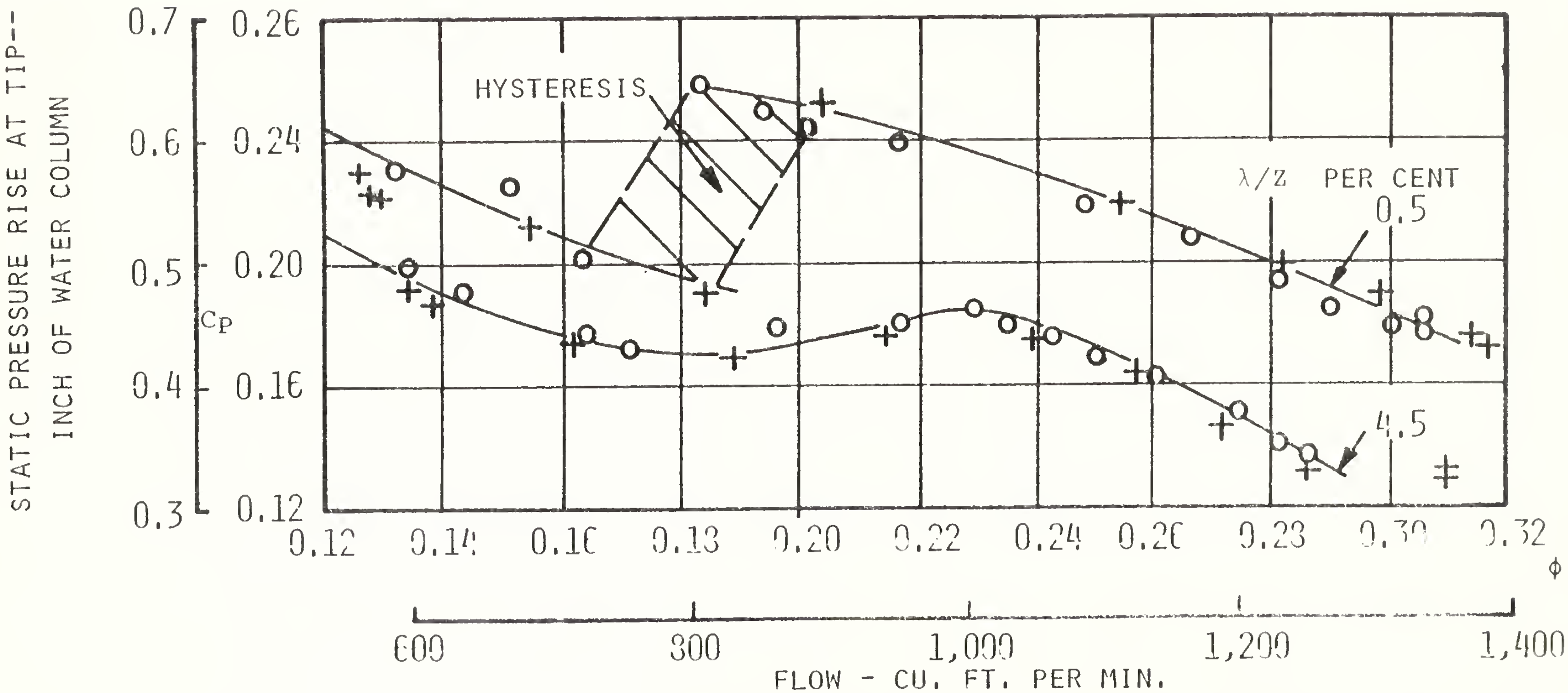


FIG. 17 EFFECT OF TIP CLEARANCE ON STALLING AND WALLPRESSURE RISE (REF. 22)

SPEED 2,000 R.P.M. + UNTHROTTLING

O THROTTLING C_P $\Delta P/\frac{1}{2}\rho U^2$

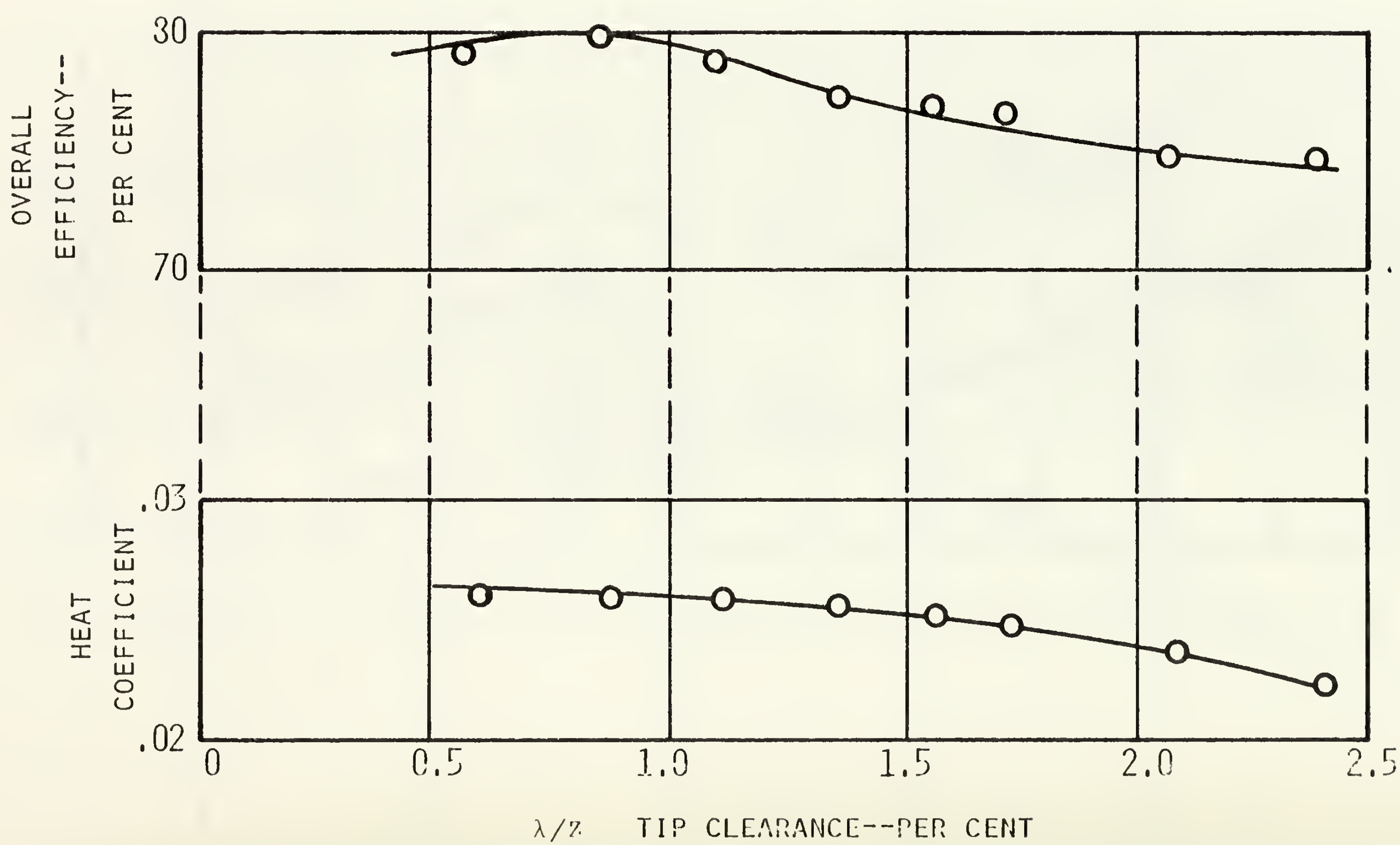


FIG. 18 EFFECT OF IMPELLER TIP CLEARANCE AT DESIGN FLOW (REF. 25)

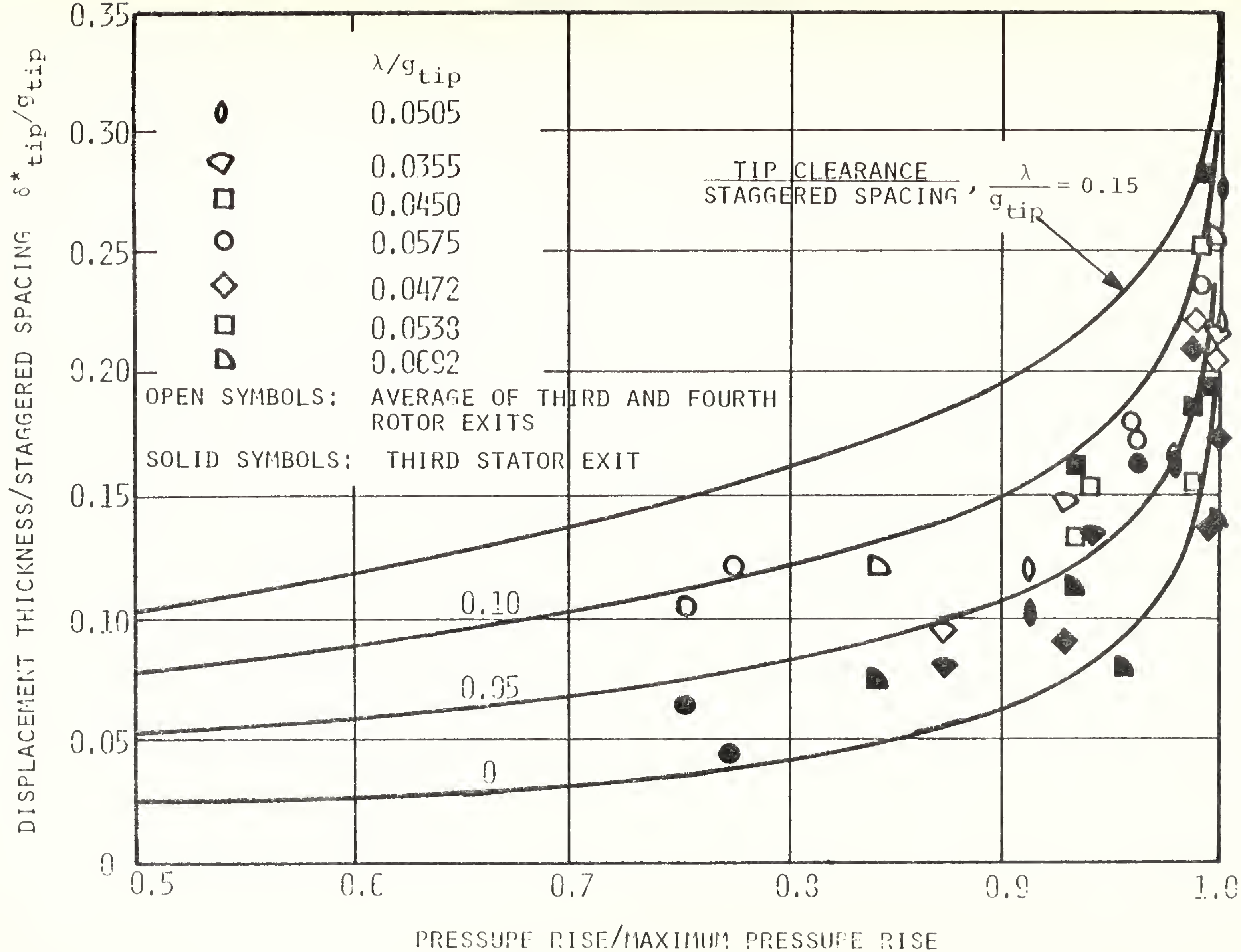


FIG. 19(A) DISPLACEMENT THICKNESSES OF CASING BOUNDARY LAYERS (FROM REF. 33)

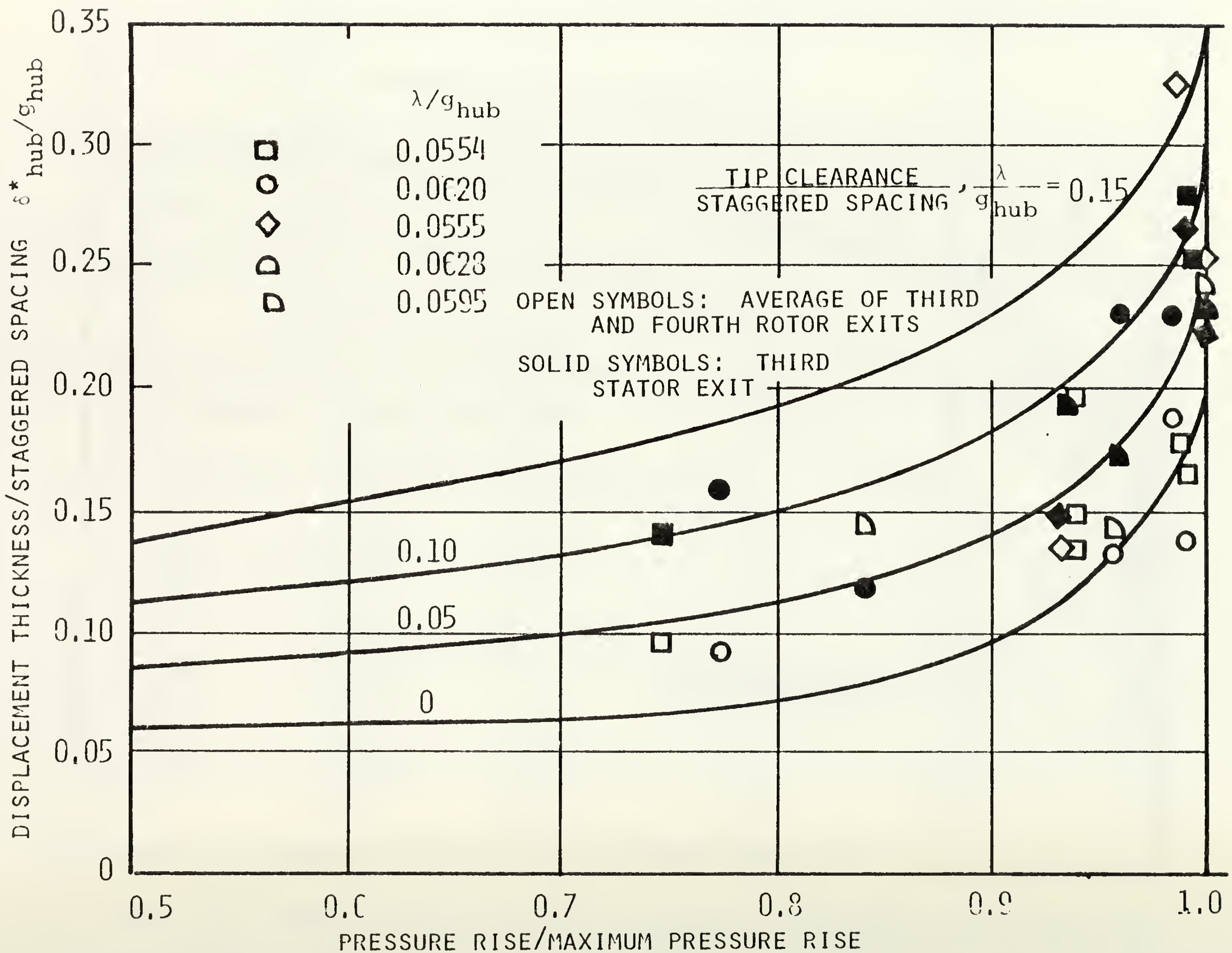


FIG. 10(B). DISPLACEMENT THICKNESSES OF CASING BOUNDARY LAYERS (FROM REF. 33)

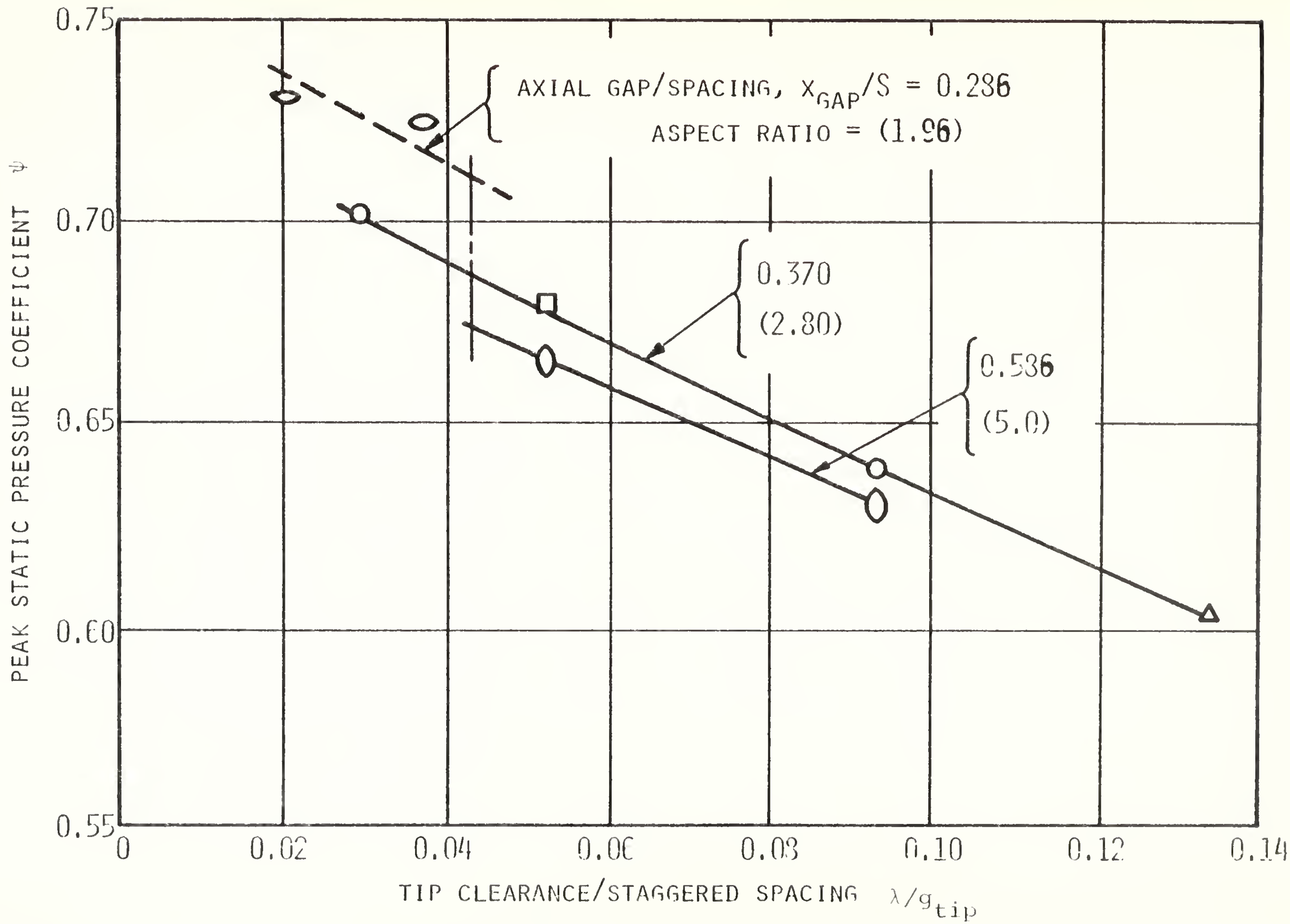


FIG.20 EFFECTS OF TIP CLEARANCE ON PEAK PRESSURE RISE (REF. 33)

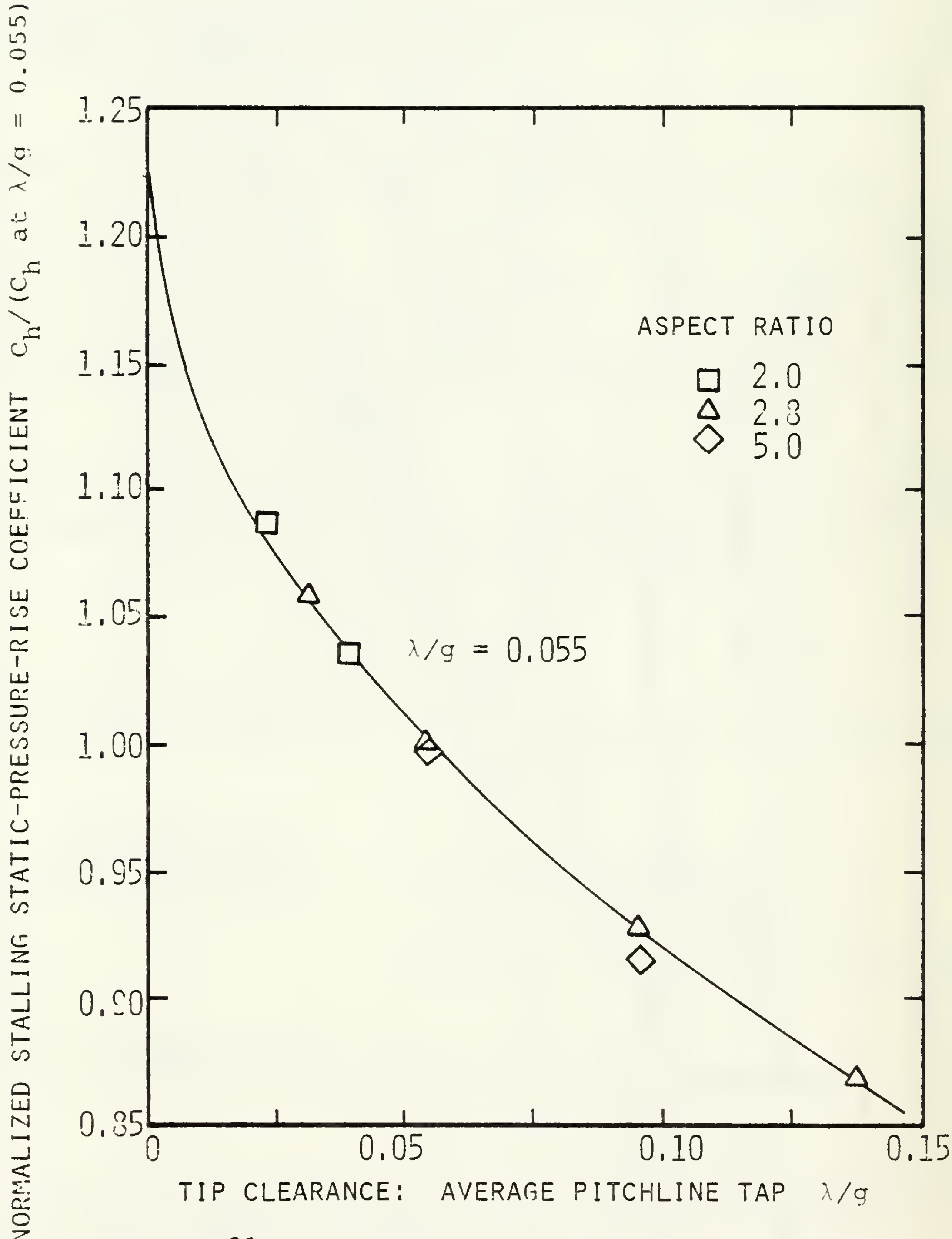


FIG. 21 EFFECT OF TIP CLEARANCE UPON NORMALISED STATIC PRESSURE RISE COEFFICIENT (REF. 34)

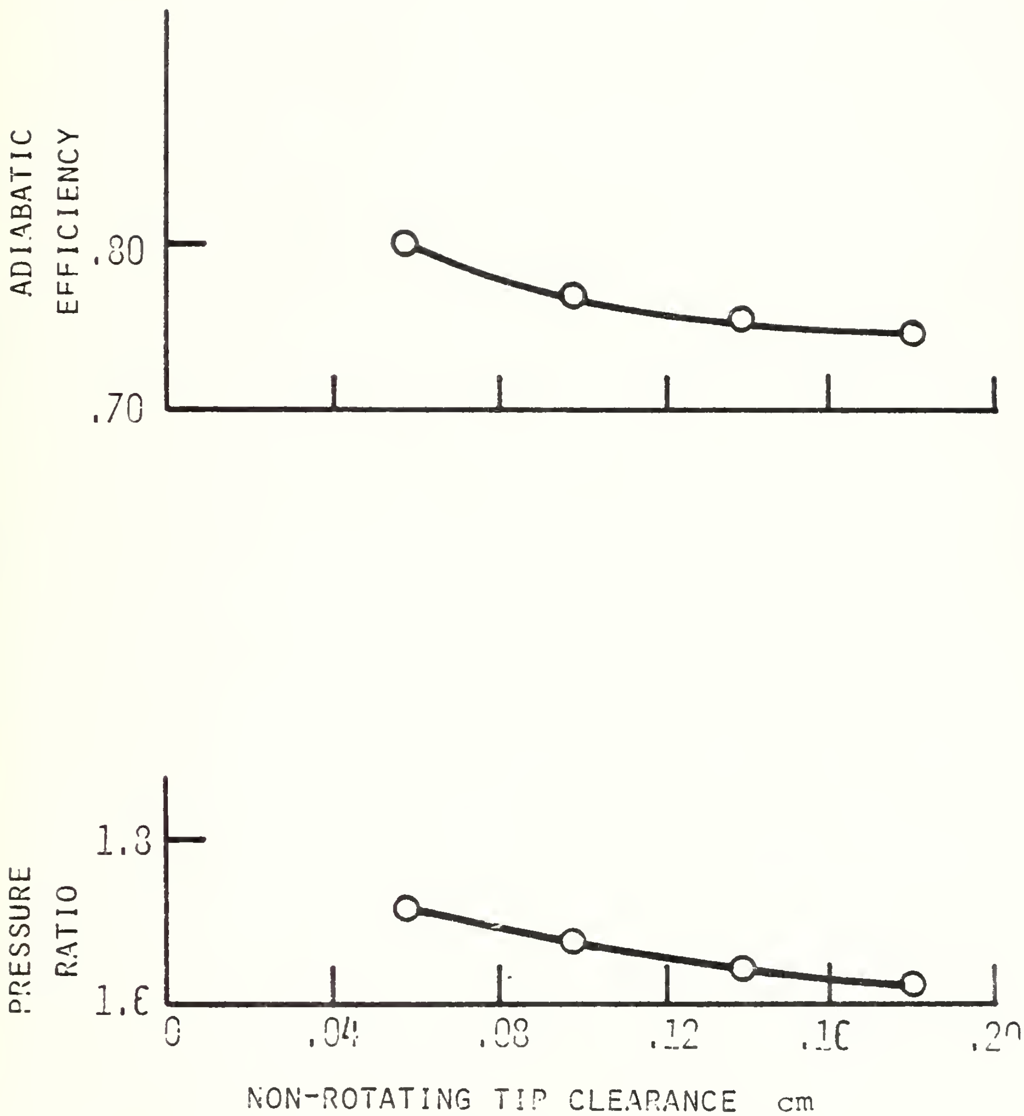


FIG. 22 EFFECT OF TIP GAP ON EFFICIENCY AND PRESSURE RATIO (REF. 35)

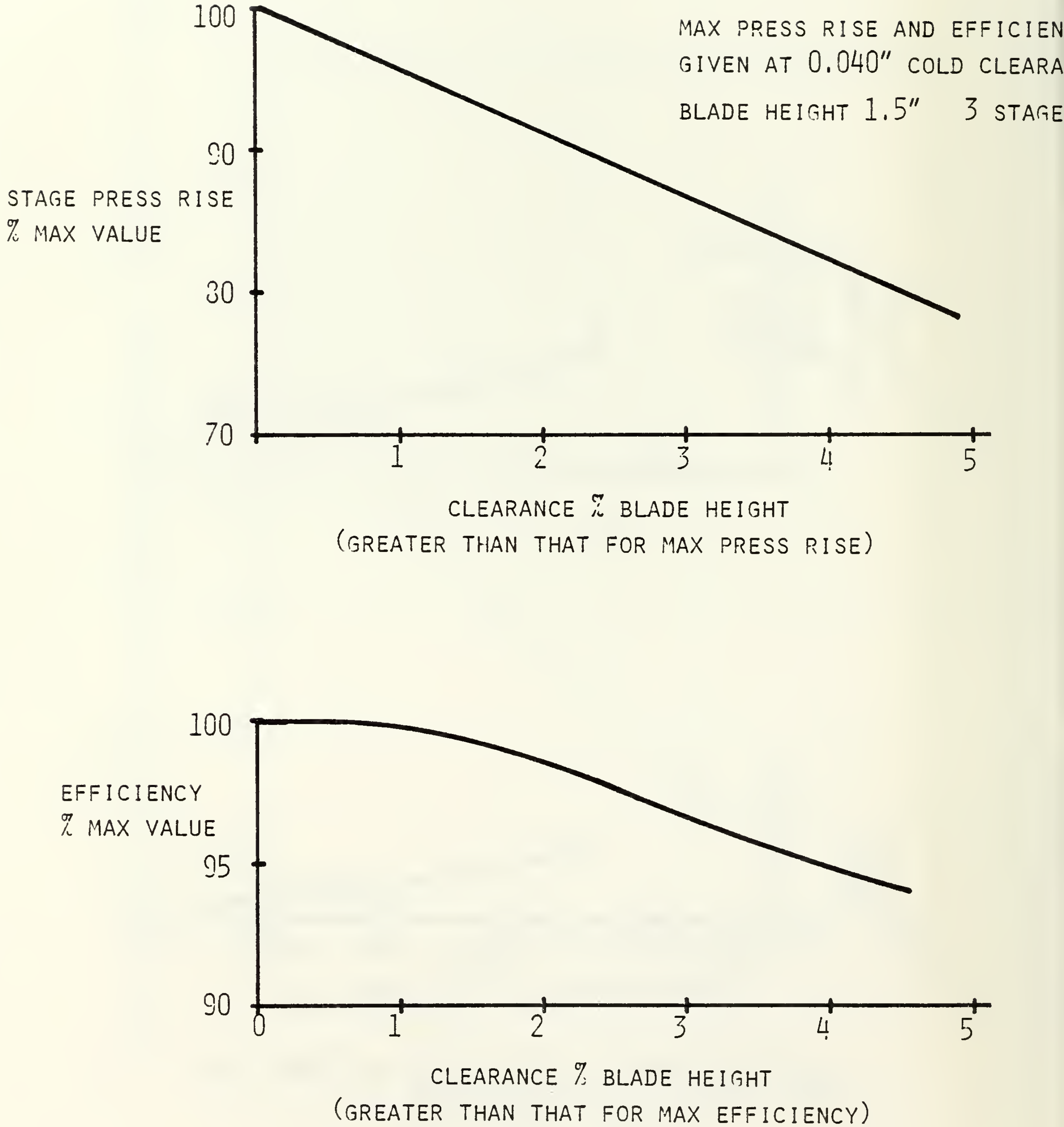


FIG. 23 EFFECT OF TIP CLEARANCE ON COMPRESSOR PERFORMANCE (REF. 37)

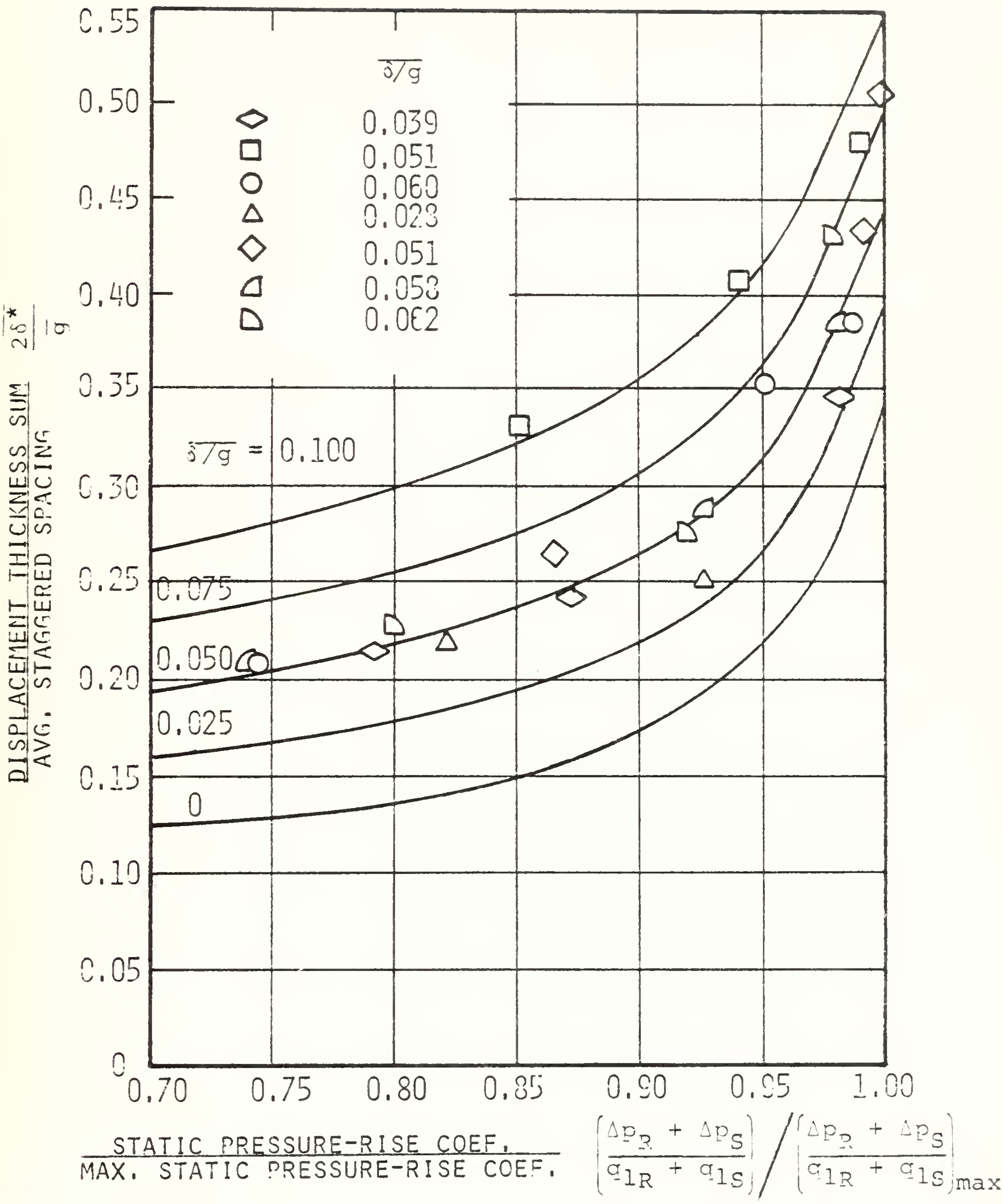
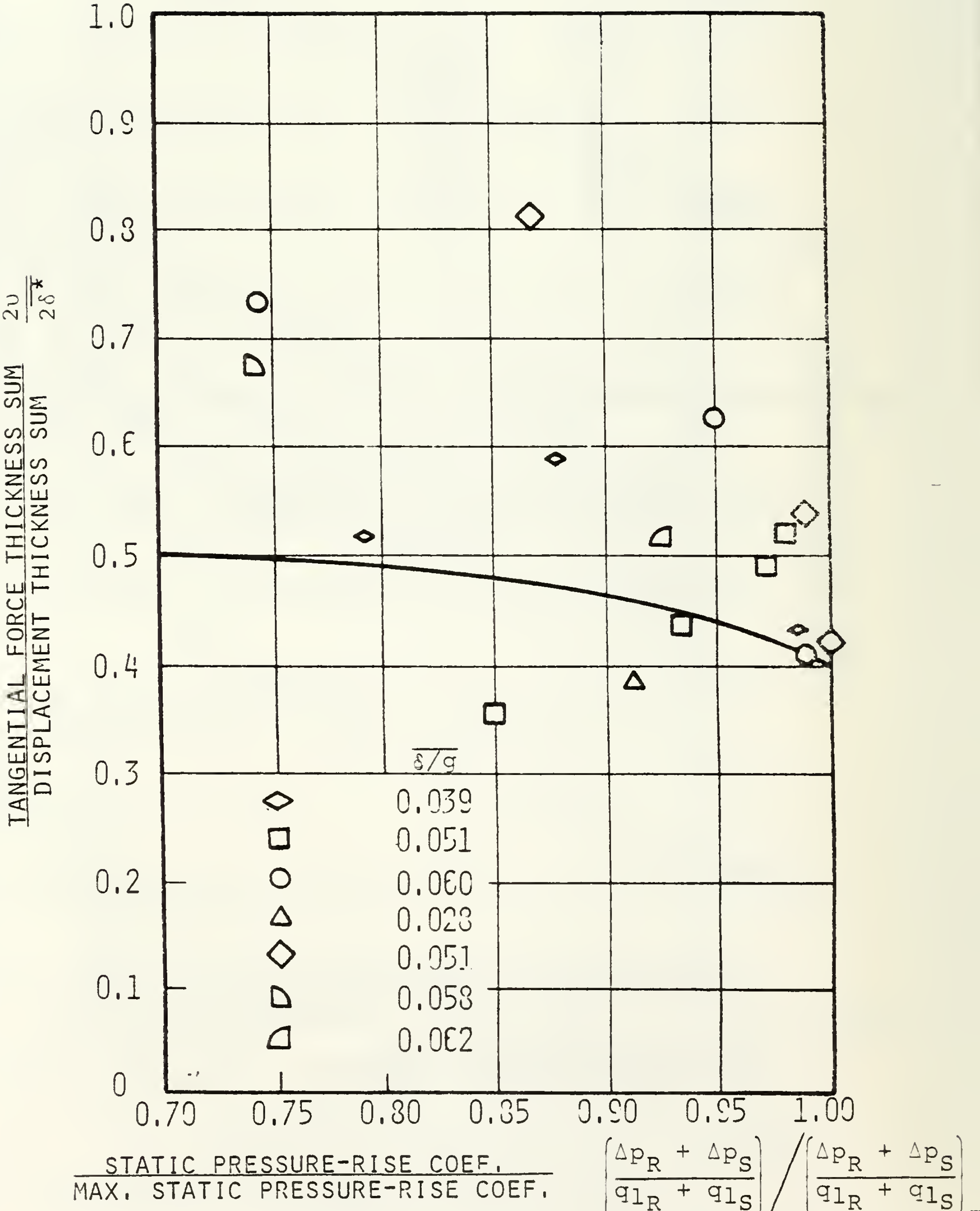


FIG. 24 SUM OF HUB AND TIP END-WALL BOUNDARY LAYER AXIAL VELOCITY DISPLACEMENT THICKNESSES (REF. 45)



$$\left(\frac{\Delta p_R + \Delta p_S}{q_{1R} + q_{1S}}\right) \Big/ \left(\frac{\Delta p_R + \Delta p_S}{q_{1R} + q_{1S}}\right)_m$$

FIG. 25 SUM OF HUB AND TIP END-WALL BOUNDARY LAYER TANGENTIAL-FORCE THICKNESSES (REF. 45)

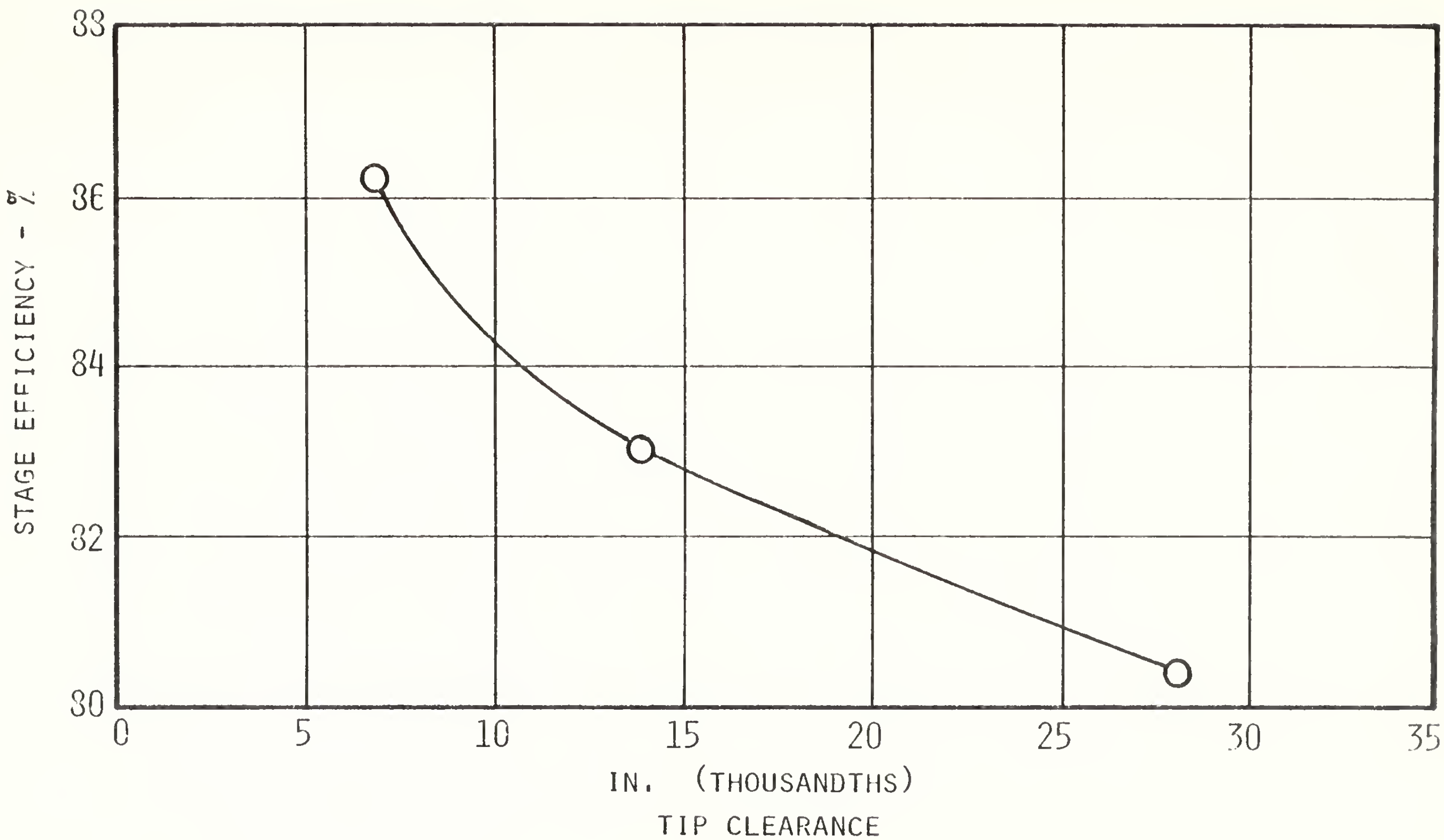


FIG. 26 EFFECT OF BLADE TIP CLEARANCE ON TURBINE STAGE EFFICIENCY (REF. 49)

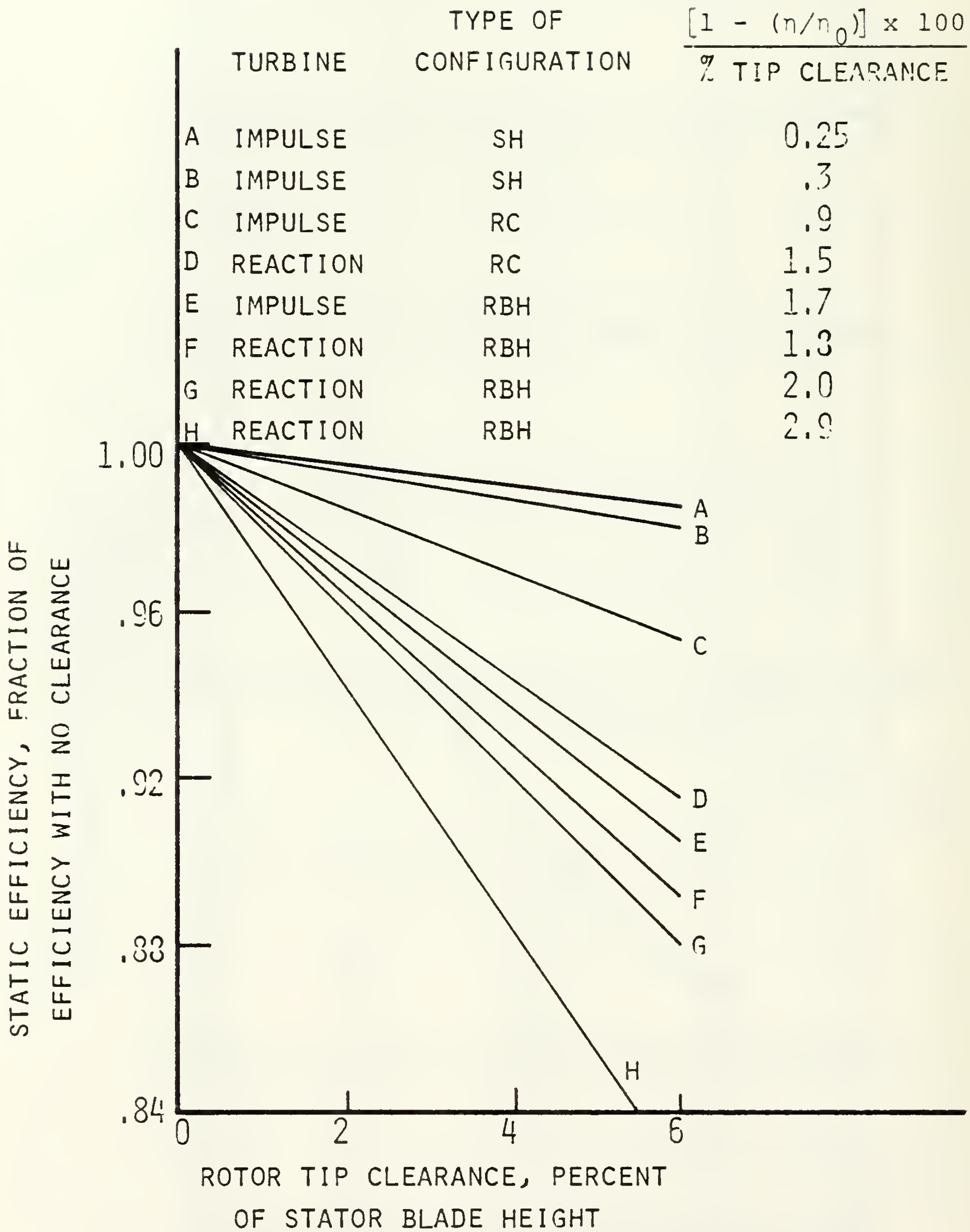


FIG. 27 EFFECT OF ROTOR TIP CLEARANCE ON PERFORMANCE FOR VARIOUS TURBINES (REF. 50)

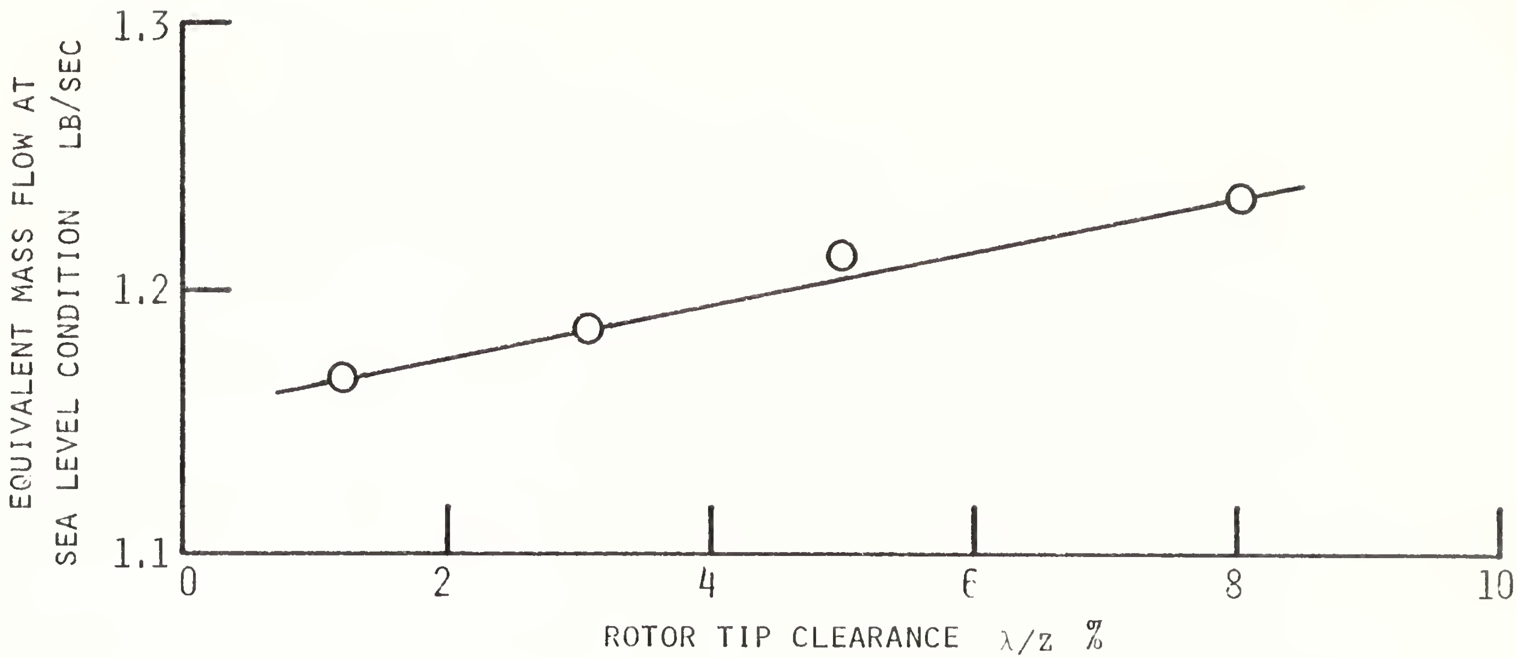


FIG. 28 EFFECT OF ROTOR TIP CLEARANCE ON EQUIVALENT MASS FLOW AT DESIGN EQUIVALENT SPEED AND PRESSURE RATIO (REF. 53)

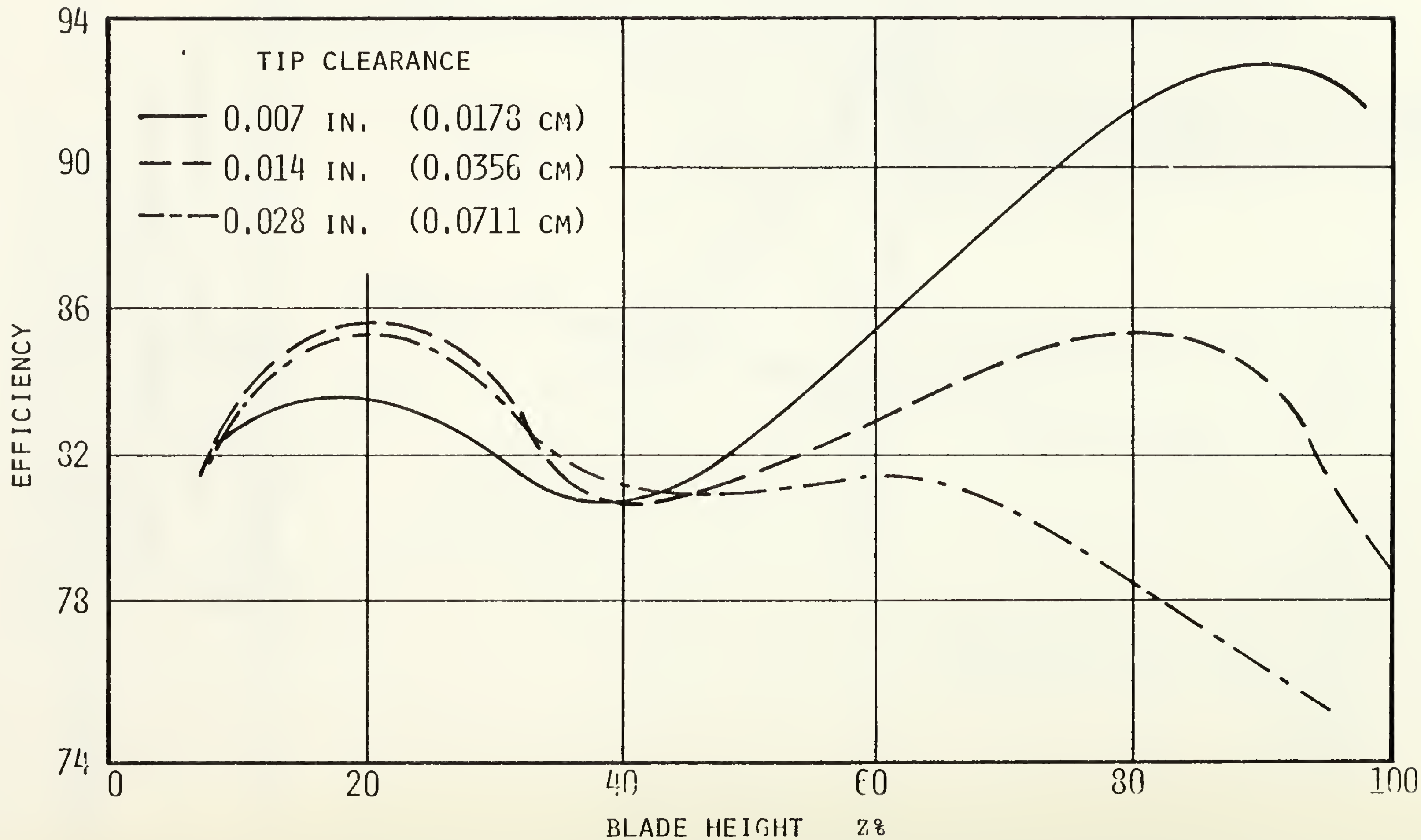


FIG. 29 EFFECT OF BLADE TIP CLEARANCE ON SPANWISE EFFICIENCY DISTRIBUTION (REF. 49)

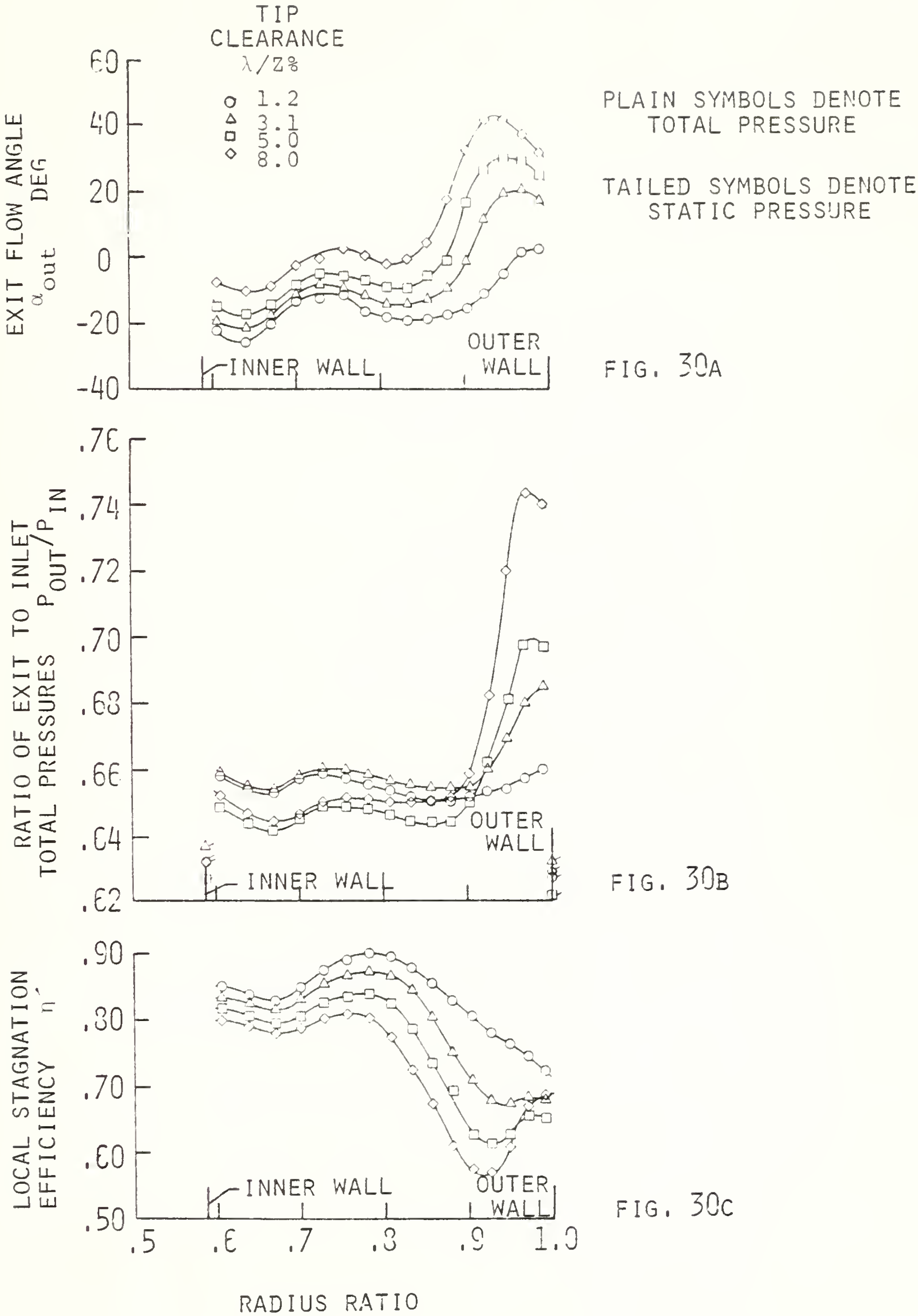


FIG. 30 SURVEY RESULTS AT ROTOR EXIT AT DESIGN EQUIVALENT SPEED AND PRESSURE RATIO (REF. 53)

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